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ELECTRO-MECHANICAL LIFT SYSTEM VERSION OF THE DT II TRACK WIDTH--ETC(U)
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ELECTRO-MECHANICAL LIFT SYSTEM VERSION
OF THE DT II TRACK WIDTH MINE PLOW
FOR M728 AND M60A1 COMBAT VEHICLES

FINAL TECHNICAL REPORT

June 1976

Prepared for

Mine Neutralization Division, Lab 1000
of the
U.S. Army Mobility Equipment
Research & Development Command
DRXFB-XN, Fort Belvoir, Virginia 22060

Contract DAAK-02-72-C-0451

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by

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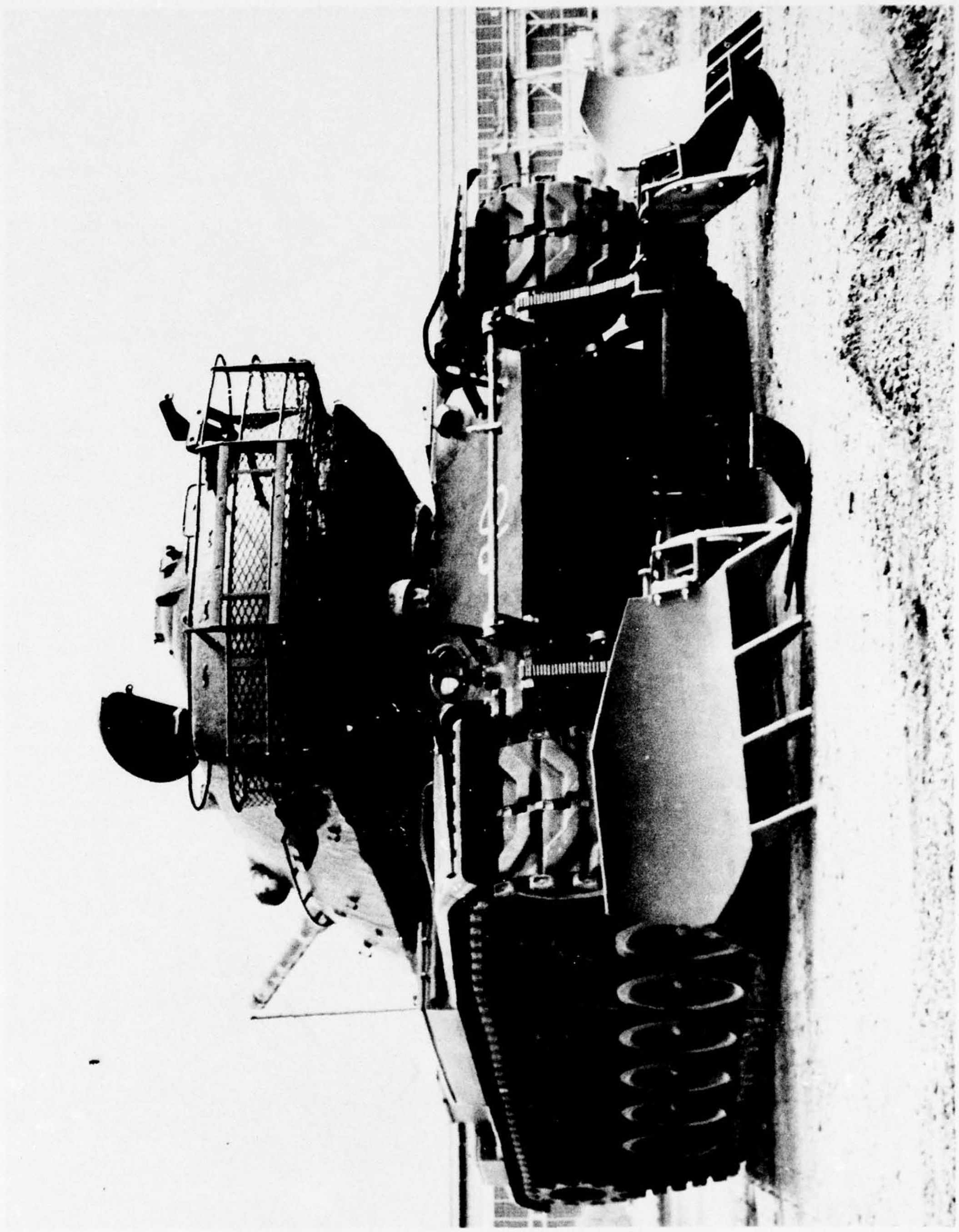

Frontispiece
DT II Track Width Mine Plow Mounted on the M60A1 Combat Vehicle

SUMMARY

This report documents the technical effort performed under contract DAAK02-72-C-0451, modifications P00012 through P00016, to resolve the major DT II Track Width Mine Plow ERP deficiencies and shortcomings assessed.

The changes addressed in this report include structural improvement along with replacement of the electro-hydraulic lift system with an electromechanical system capable of being quickly mounted to a vehicle. USAMERADCOM testing at Aberdeen Proving Grounds during the months of March and April 1976 of an M60A1 vehicle equipped with a Track Width Mine Plow System so modified, has demonstrated the feasibility of the improved system.

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1.0 INTRODUCTION

The DT II mine plow is a device installed on the front of an M60A1 or M728 combat vehicle hull to remove land mines from the path of the vehicle tracks and detonate tilt rod mines that lie between the paths cleared by the plows. A mine plow system is composed of three major subsystems: a plow subsystem, an electrical subsystem, and a hydraulic subsystem.

The structural subsystem consists of two plowing units, one for each side, that are pushed ahead of the vehicle tracks to clear a path the width of each track plus a few additional inches to either side. A chain curtain is suspended between the plowing units to detonate mechanically tripped (tilt rod) mines which might otherwise be missed because they lie between the paths cleared by the plows. Major plow components for each side of the vehicle are a push-beam, a moldboard with four downward projecting scarifier teeth, and a skid shoe. The scarifier teeth penetrate the earth to dislodge buried mines and bring them to the surface where they are shunted aside by the moldboard. The skidshoe controls the depth of penetration of the scarifier teeth and reacts to terrain undulation. The moldboard and skidshoe are installed on one end of the pushbeam; the other end of the pushbeam pivots against a plow mounting bracket which is secured to the vehicle hull. When plowing, each plowing unit operates independently of the other, freely following the natural contours of the terrain it encounters. Lifting chains are installed between the pushbeams and the plow mounting bracket. These chains, together with hydraulic cylinders installed inside the pushbeams, are used to raise the plows to the transport position or to lower them to the plowing (or operating) position. In the transport position, a travel lock assembly on the vehicle hull holds the plows securely in place.

The electrical subsystem consists of wiring harnesses, a junction box, a control box with raise/lower and on/off switches, a power on indicator light, and a special headlight adapter assembly for the tank. The electrical system activates and deactivates the hydraulic system.

The hydraulic subsystem furnishes hydraulic pressure to actuate the hydraulic cylinders which raise or lower the plows. The system consists of a hydraulic reservoir; a pump and drive unit composed of a pump that is driven through a magnetic clutch by a right angle drive transmission power take off; and associated tubes, hoses, valves, fittings, and other hydraulic system components.

Results of Development Test II (Engineering and Service phase) testing of the tank-mounted mine clearing plow device indicate certain design shortcomings and deficiencies, each affecting the plowing mission in various degrees.

Chrysler, under Contract DAAK-02-72-C-0451 was to design, document, and fabricate hardware; correcting three general categories of shortcomings/ deficiencies: The leakage-prone hydraulic lift system, the lifting chain which rusted and allowed the system within the pushbeam to be jammed with debris; and structural failures of the pushbeam and the mounting bracket. This report describes an expedited solution to these problems.

An electromechanical ball screw actuator and control system was designed to replace the entire hydraulic lift and control system, thereby correcting the leakage problem and providing a system that can be quickly mounted to all M60A1 vehicles. A boot was added to the existing chain to provide weather and debris protection. Structural beef-up was added where necessary, with a complete redesign of the center clevis unit. These changes had to be functionally integrated into the total TWMP system and consequently they affected all parts of the TWMP (except the moldboard and skid shoe) which in turn necessitated various design compromises.

The specific designs described herein are the final result of early extensive preliminary investigations. These preliminary approaches were discussed with MERADCOM and with their concurrence, lead to the selection of the design concepts which were reduced to hardware.

2.0 SYSTEM REQUIREMENTS

The system functional requirements to which the deficiency corrections were to be designed were established from the contract work statements and through the results of several design review meetings held during the spring and summer of 1975. In summary the general system requirement for the Chrysler proposed new Electromechanical Actuator Lift System (E-MALS) dictated that the original TWMP operational capabilities would not be degraded, and would be enhanced wherever possible within the constraint of not altering the basic TWMP configuration. The specific E-MALS functional requirements are detailed in Section 4.1.

In certain areas, some structural modifications were made in the field during DT-II testing. CDD was to investigate these modifications, revise the pertinent drawings and correct the modifications, with new documentation, as necessary. Since no plowing loads were known, this task required that CDD generate a TWMP loads criteria as input to a NASTRAN modelling of the TWMP structural members to provide a base for structural changes.

3.0 DESCRIPTION - OPERATION ELECTRO-MECHANICAL ACTUATOR LIFT SYSTEM

The electro-hydraulic system to raise and lower the plow has been replaced by an Electro-Mechanical Actuator Lift System (E-MALS) which essentially performs the same functions. This section will describe this change. The description of the changes to the chain system, mounting bracket, pushbeam, and center clevis are discussed in their respective sections.

The E-MALS consists of three basic subsystems: The actuator unit (Figure 1), the control panel (Figures 2 and 3), and the electrical harnesses. The actuator unit is an integrated, one-piece assembly made to fit inside the pushbeam cavity to replace the hydraulic cylinder. The large chain roller assembly is removed from the hydraulic cylinder piston and is attached to the actuator unit to perform its same function. The actuator unit consists of a housing weldment which contains an electric drive motor and a speed reduction gearbox (Figure 4) whose output drives a ball bearing screw linearly in and out the same manner as the hydraulic piston.

The control panel is a box weldment that contains most of the electronic control circuitry, which signals the actuator motor to perform. A few control components are located in proximity with the motor in the actuator unit due to their functional requirements. The control box is mounted in the driver's compartment using an adapter mounting bracket at a position above the driver's control panel. This mounting bracket also mounts additional control components (ballast resistors). The control panel face contains all the operational switches and indicator lamps arranged and designed to meet human factors specifications.

The electrical harnesses are routed to receive vehicle power from the vehicle's slave start receptacle, to enter and leave the control box and then to exit the hull through the vehicle's two headlamp adaptors (Figure 5) to feed each of the actuators. Outside the hull, the harnesses are routed through the mounting bracket, for protection, and into the pushbeam. Connectors and terminal boards are used throughout the harness runs for convenient disconnects for the TWMP installation. The selection of the control box mounting and location and the harness routings and terminations were made to accommodate an expedient installation on an existing M60A1 vehicle.

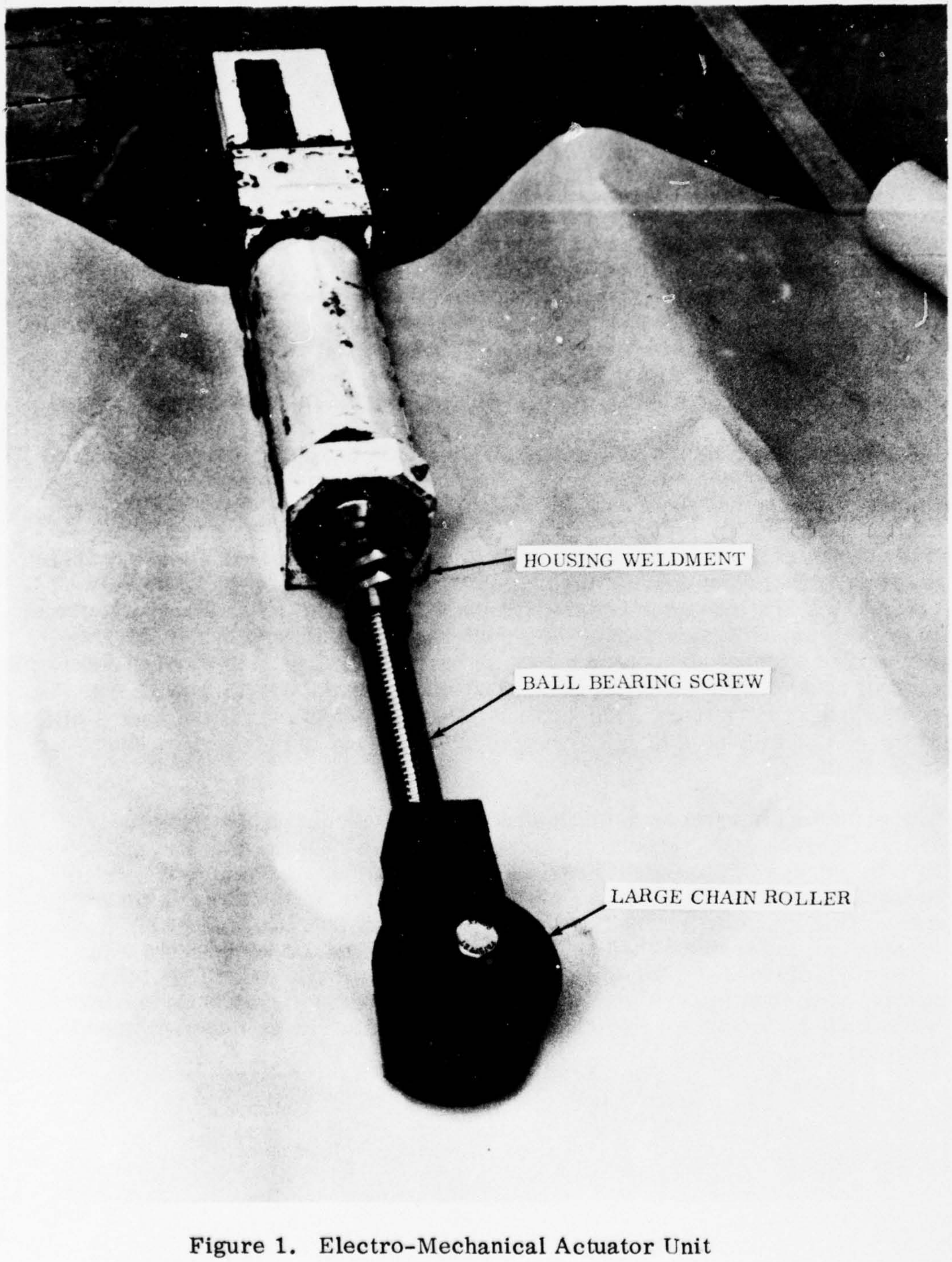

Figure 1. Electro-Mechanical Actuator Unit

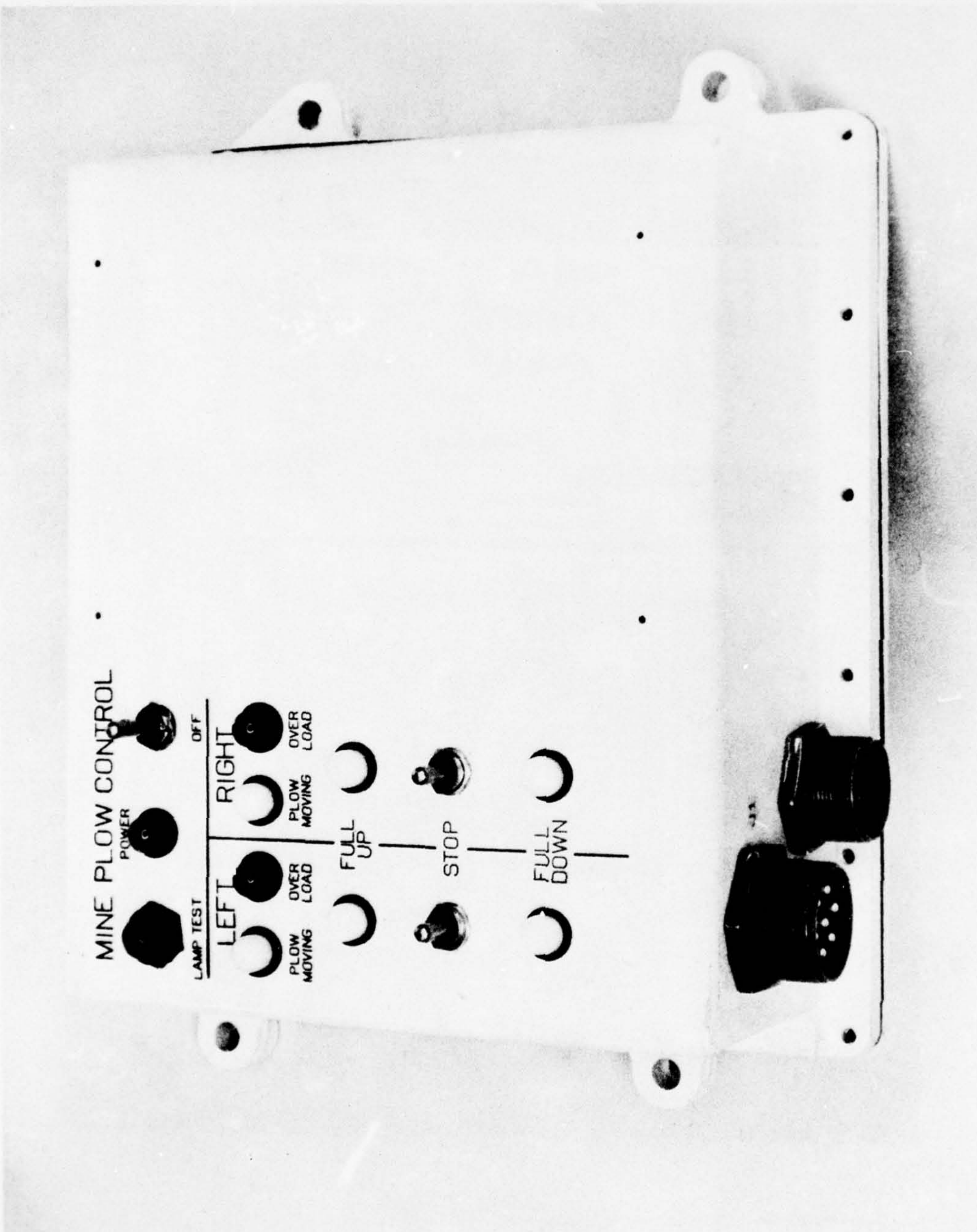

Figure 2. Control Panel (Outside)

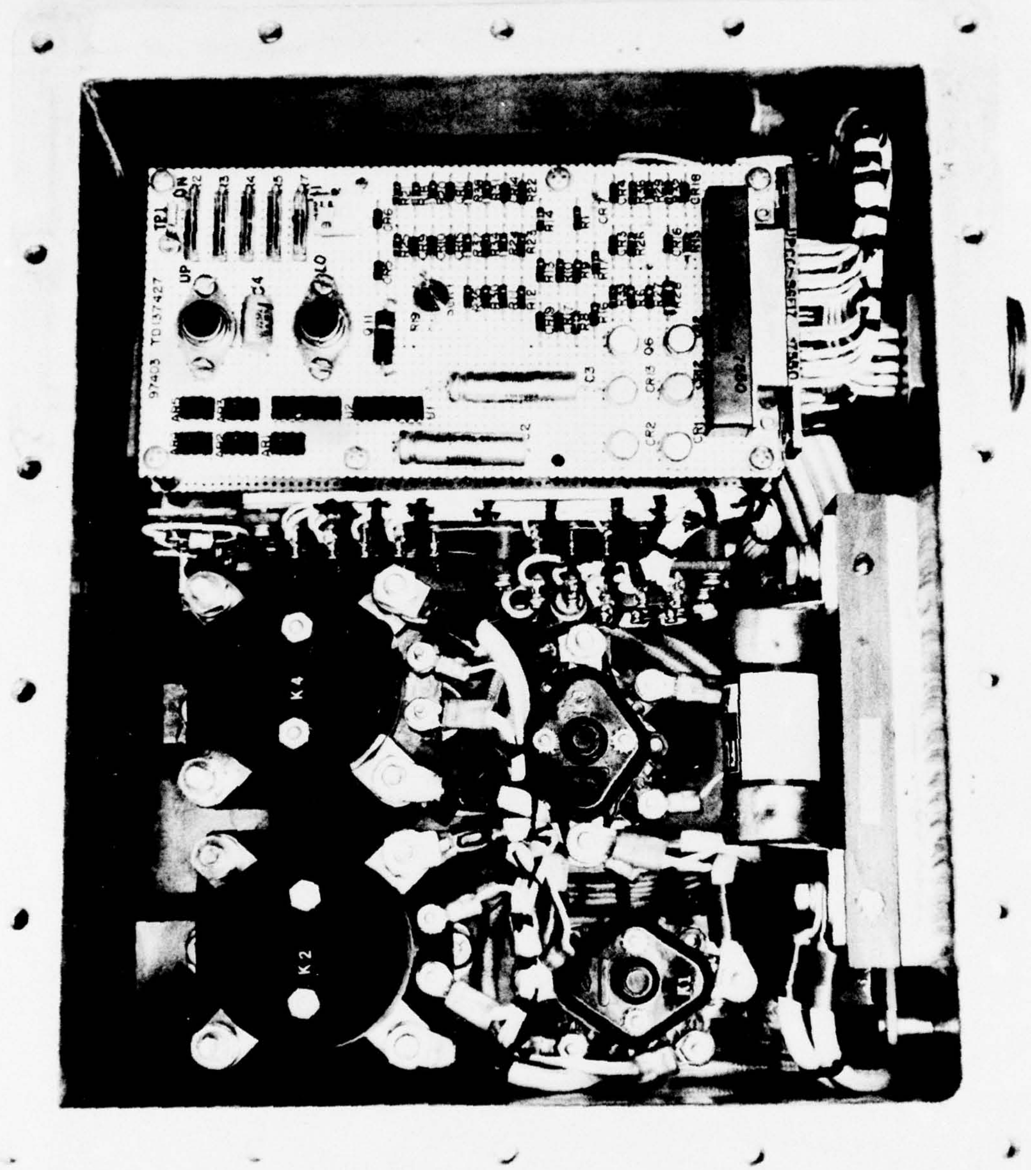

Figure 3. Control Panel (Inside).

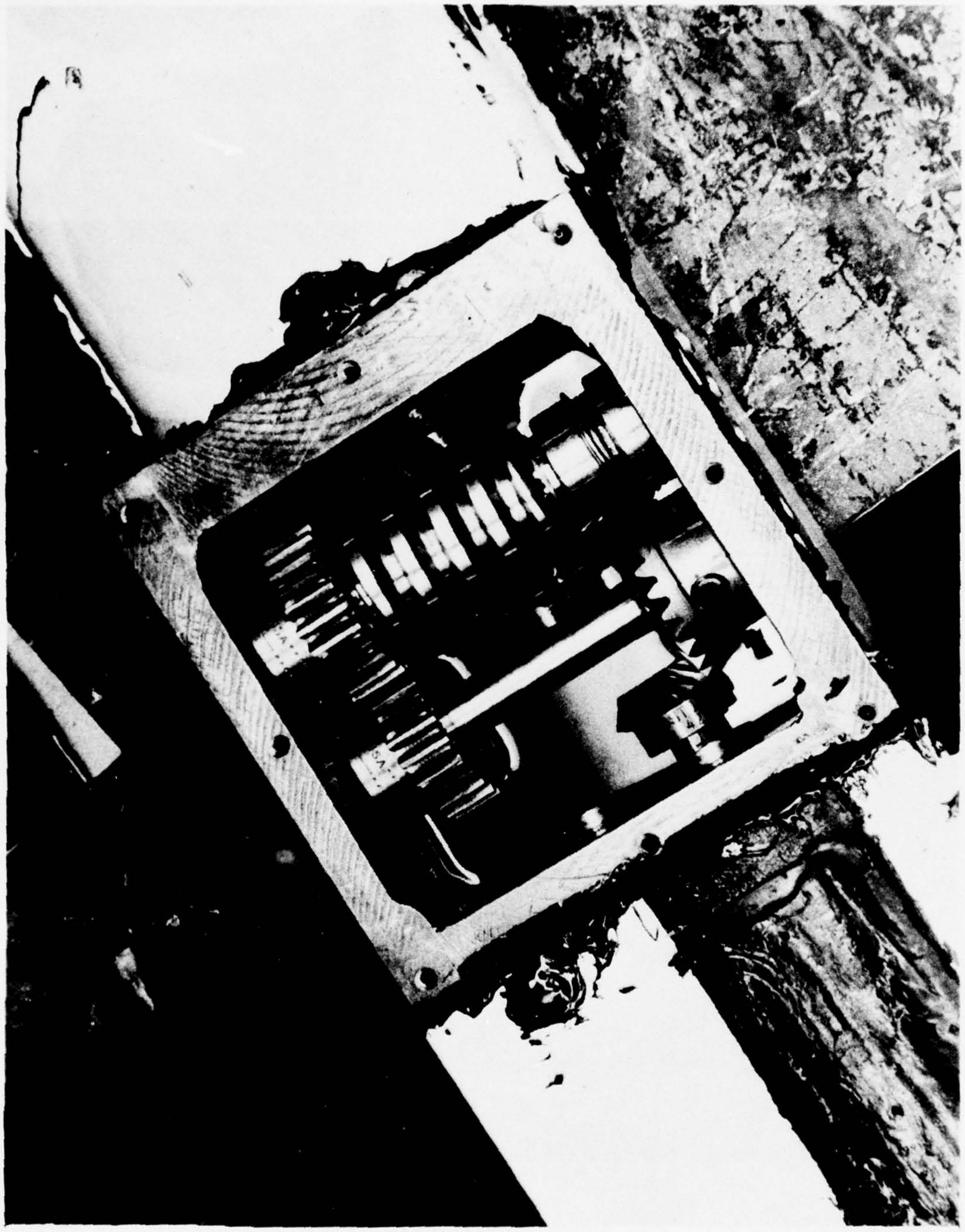

Figure 4. Speed Reduction Gearbox

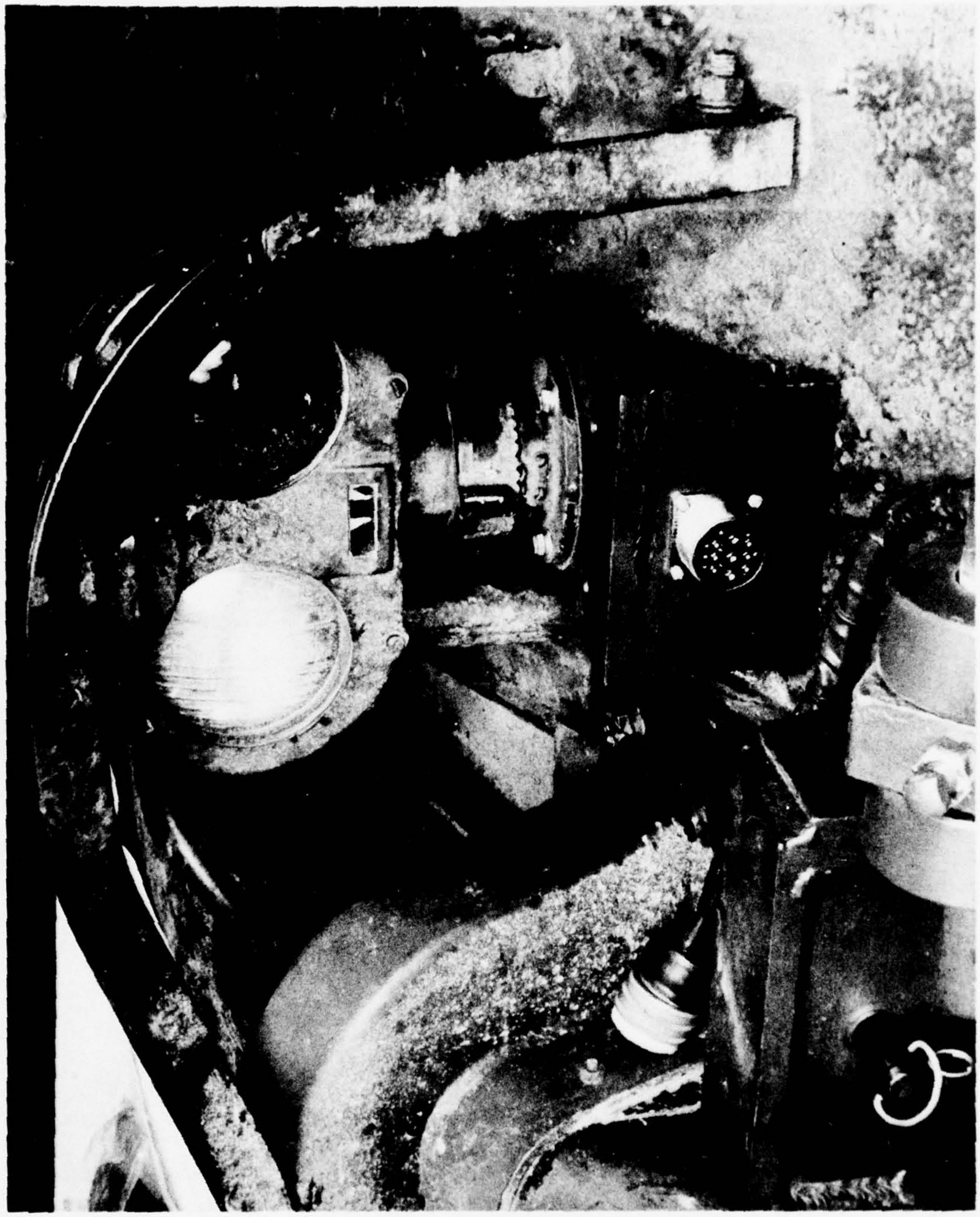

Figure 5. Headlamp Fittings

A diagram of the control panel and a summary of the E-MALS basic control operation are given in appendix H. The E-MALS is designed to allow independent operation of each plow, while plowing or not plowing, or with vehicle engine operating of not operating. To raise the plow, the driver activates the raise switch and raising takes place automatically. The raised position is indicated to the driver by a panel lamp and he may then proceed to manually latch the plow. Sufficient reverse torque action of the gearbox is present to prevent a dirt loaded plow from descending under its own weight. After the plow is latched, the switch is placed in the center position. To lower the latched plow, the driver moves the raising switch. The circuit causes the plow to raise allowing manual unlatching then the switch may be reversed to the lower motion. The lowering motion will continue to a preset actuator screw travel before automatic cutoff. This means that the full preset length of lift chain will be out regardless at what position the plow comes to rest. Indicator lamps are included to show that the plow is in motion, that the plow is stationary in either the up or down position, and if an overload occurs on the motor due to exceeding the design duty cycle. To resume plowing the overload must first be removed (for example, reversing the vehicle) and the control system reset by placing the raising switch first into the center position.

During the in-house functional and qualifying tests an additional override switch and circuit was added to the control box as an expediency measure to revise certain functions that were originally designed into the control circuit. The description and operation of this addition are given in section 6.0.

4.0 DESIGN ANALYSIS

The objectives of the design analysis effort were to obtain workable solutions to the DT-II deficiencies prior to the scheduled DEVA IPR. This meant that any changes to the TWMP system had to be minimal, functional, and capable of adaption into the TWMP development cycle without repeating the DT-II test phase. The approach then was to determine what caused the deficiency and to correct the existing design using proven methods. This approach posed no problems with the structural member or rusty chain deficiencies. However, the oil leakage problem in the hydraulic lift system, though simple to correct in itself, was conditioned by another factor which was not a deficiency. That factor was the high retrofit cost of the add-on hydraulic system required for the M60A1 vehicle. Thus, the design analysis of the lift system concerned the replacement of an entire system but still under the desired objectives of a simple, quick, low cost replacement with minimal changes to the TWMP hardware.

4.1 ELECTROMECHANICAL ACTUATOR LIFT SYSTEM (E-MALS)

An electromechanical plow lifting system was designed as a direct replacement for the electrohydraulic lifting system. The original desired design directive was to produce a formal data package that would provide the details to procure a system which would, in essence, be designated as a proven concept suitable for DT-III testing. This objective was energetic in that the development time associated with a normal engineering development cycle had to be eliminated and dependence put entirely on analytical proof of system functional operation. This approach was ameliorated by the opportunity to correct any design shortcomings through the fabrication and functional test on a tank, of a concept feasibility demonstration model of the E-MALS. Much of the E-MALS hardware described in the ensuing sections pertain to this demonstration model but other than size and performance rating differences imposed due to using GFE pushbeams from DT-II tests, the E-MALS concept is the same.

The E-MALS lifting system was designed according to a set of performance requirements embodied in a force-time duty cycle which, lacking specific plowing loads, was constructed to match the capabilities of the electrohydraulic lifting system. The nominal plow lifting time was set at 15 to 30 seconds. Another requirement was that each plow is to be activated independent of each other. Also, the design adhered to the requirements of field maintenance replacement of the system, the ambient operational temperature extremes of -40 to +125°F, the environment of mud, ice, and water, simple foolproof and adjustable operator controls, and minimal revisions to the M60A1 vehicle. The electrical system power usage was designed to not degrade the vehicle's power requirements. The electrical harnesses required had to be independent of the vehicle's harness system and readily installable.

The control system functional requirements grew more sophisticated as the E-MALS system evolved. Built-in adjustment, overload, safety, and simplistic operation features were deemed desirous. The resultant principal design requirements of the control system were set as to:

1. Provide simple but fail safe operation
2. Permit independent and accurate position stops
3. Provide simple and clear indications
4. Permit single or combined (2 plow) operation
5. Permit dual speed - torque operation (more torque to raise than to lower)
6. Control overtravel and bind up
7. Provide flexibility during design-development

4.1.1 Design Approach

The basic design approach for the mechanical actuator was to select an electric motor driven ball bearing screw mechanism (BBSM) which translates rotational motion of a threaded screw shaft to a linear motion thereby simulating the piston motion of the hydraulic cylinder actuator. With a necessary gearbox to reduce the motor speed, this assembly of motor, gearbox, and BBSM was to be packaged as a unit for the feasibility demonstration model since it had to be inserted into the DT II plow pushbeam's only access at the forward end. The control system completes the total system and is comprised of a driver compartment mounted control box and ballast resistors and harness assemblies which provide control and power to the outside actuator lifting assembly in the plow. The vehicle's headlamp fittings were selected as the most expedient routing for the harness out of the hull.

Commercial catalog hardware was used whenever possible. Those selected major items such as the BBSM, gears, and motor were discussed with the manufacturer for performance, cost, and availability. To avoid long lead procurement times those items which had to be fabricated such as support structures, housings, environmental shields, vehicle interface fittings, and the control box were initially detailed as weldments for this demonstration model. The gearbox housing and the control box housing should be specified as castings for any production version.

The electrical control circuits, selected for proper operation of the E-MALS system are hybrid and are comprised of semiconductor, integrated circuits and electromechanical components as are best suited to perform their intended functions. It was not found practical, primarily because of cost, to develop an all solid-state control with the relatively high current requirements. The approach to the control system design included considerations of well established maintenance procedures. The electrical circuits outside the vehicle hull have been restricted to the use of motors, relays, wiring, etc. The more complex circuits are located within the sealed control box, and on two plug-in circuit boards within the box. Major portions of the circuitry may be simply exchanged by plug in, with more detail repair on the board assembly properly done at a depot.

4.1.2 Duty Cycle

The duty cycle is defined as the raising design load variation on the BBSM screw as a function of time for a selected set of conditions of plow operations. The loading during plow lowering was not considered as critical as raising and was not analyzed. Basically the duty cycle is comprised of two phases; "dynamic" which raises the plow out of the ground while plowing, and "static" where the plow is raised from above the ground to its stowed position.

Dynamic loads occur when the driver starts to raise the plow while the vehicle is moving and the plow is plowing. While plowing, the skid shoe resists all the vertical "digging" load of the plow until the limp lift chain has shortened to the point where it lifts the skid shoe off the ground. At this moment the lift chain now takes over the vertical plow loads and this load is directly felt by the BBSM screw as axial compression. The dynamic load reduces to zero as the plow disengages from the ground.

Lacking any specific ground breaking load data, the lifting requirements were selected to match the capability of the present hydraulic cylinder of 25,133 pounds, consequently the maximum design compression lift load on the ball screw is limited by the control system to this load. This value was derived from the hydraulic piston area and the regulated pressure which is 2,000 psig.

The static load is derived from the combined dead weight of the plow which is presumed to remain at 1,463 pounds plus a selected 400 pounds of dirt which is considered to adhere the plow. This static load is also considered to be present during the dynamic lift phase. Note that the resultant duty cycle is a combination of an actual physical load (static) and a rated value of a hydraulic cylinder (dynamic).

Operationally, the plow is designed to plow to a depth of 9 inches as the vehicle traverses over hills and through dips. This is allowed by sufficient slack in the lift chain. From the lift chain length, it has been determined from a geometric layout that the plow can lower approximately an additional 9 1/2 degrees before the chain becomes taut. This 9 1/2 degrees has been defined in this analysis as the variation between level plowing, over the hill plowing and through dip plowing and represents 2.775 inches of ball screw travel. Also, this 2.775 inches represents the BBSM screw travel during dynamic loading (breaking ground). Figure 6 depicts the geometric position of the plow pushbeam and the yoke roller point for the three plowing attitudes. Note that at the maximum chain extension, the ball screw must travel 2.775 inches to get the plow out of the ground leaving 13.225 inches to raise it to the stowed position; and at level ground, the ball screw must travel 2.775 inches to take up the lift chain slack then another 2.775 inches to raise the plow out of the ground, leaving 10.45 inches to raise it to the stowed position. Similarly at the highest plowing condition, the BBSM screw must take up 2 x 2.775 = 5.55 inches of lift chain slack, then another 2.775 inches to take the plow out of the ground leaving 7.675 inches to raise it to the stowed position. From this scenario it can be seen that the dynamic load phase is the same for any plowing condition, i.e., the ball screw travels 2.775 inches to lift the plow out of the ground. The static load phase, however, is different in that the BBSM screw travels variable distances to lift the static load. This means that with the same actuator motor input performance and plow dirt load the static lift travel times will vary with the initial plowing attitude. Figure 7 shows the force distribution of the lifting load and the assumed horizontal hinge point radial moment arms of the plow dead weight and the 400 lb dirt weight. Note that as the plow is raised the static moment arms vary and the maximum lift load occurs when the pushbeam is horizontal. The axial compression force (F) on the BBSM screw is seen to be twice the chain tension load (W_c). From a scaled geometric layout of the three plow positions, the static loads were computed as shown on table I. For the design duty cycle, the 400 lb dirt load is selected giving the maximum static load of 4,658 at the beginning of static lifting and 3,916 lbs at the stowed position.

TABLE I. STATIC LIFT LOAD, F, ON BBSM SCREW

Dirt Weight	Plow Position: Max Down	Level Ground	Stowed Up
0	3,388	3,512	2,950
400 lbs	4,494	4,658	3,916
800 lbs	5,600	5,806	4,882

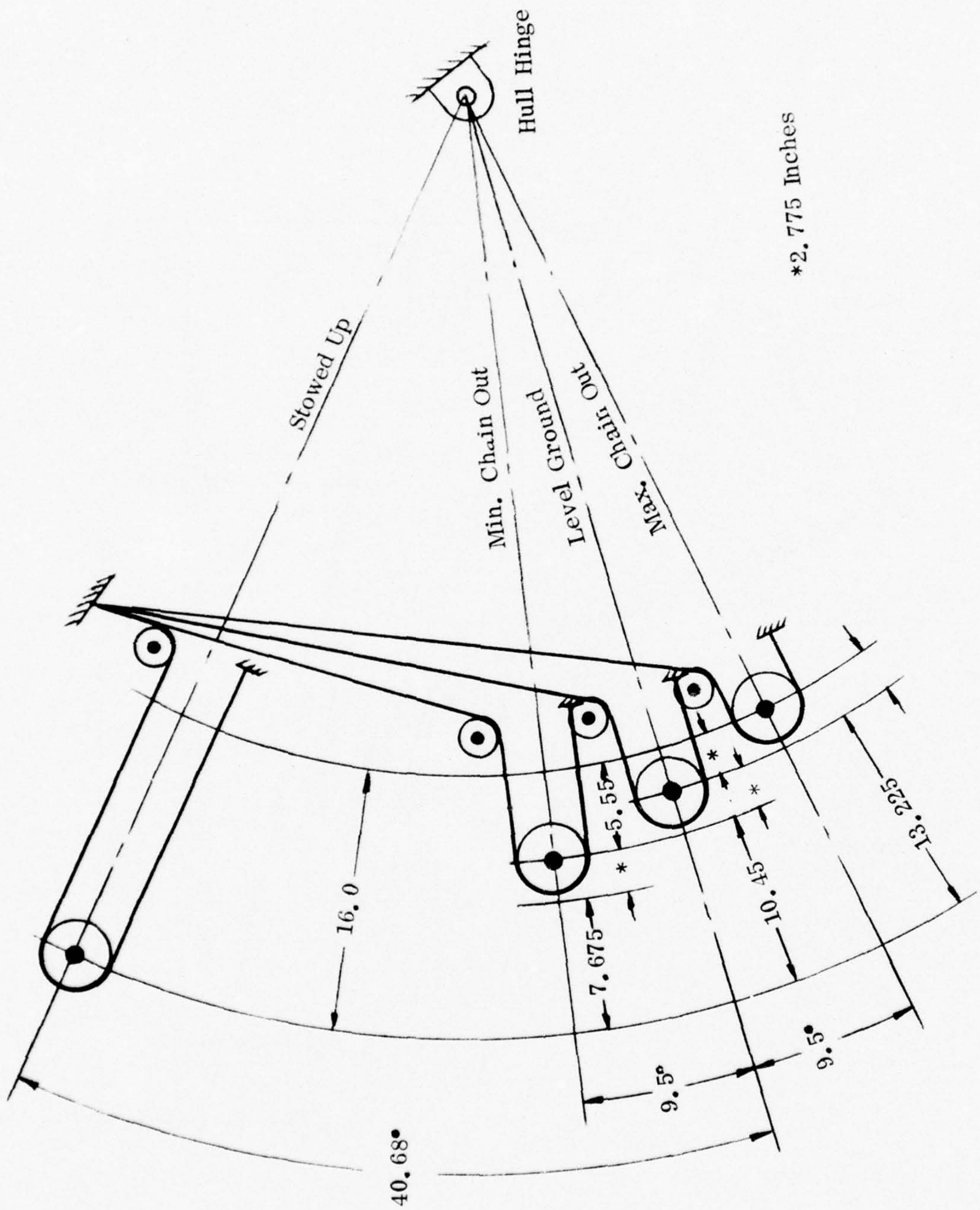

Figure 6. Plow Position Geometry

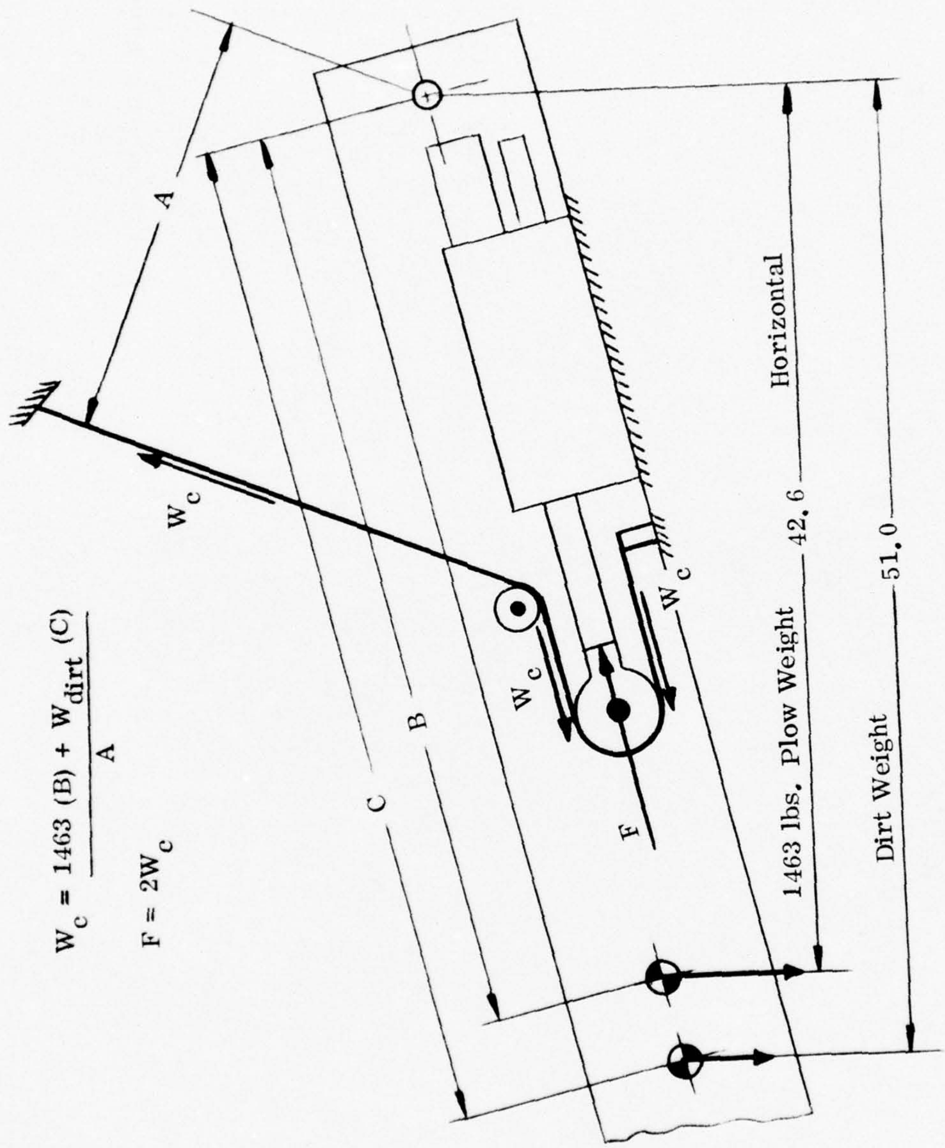

Figure 7. Static Lifting Force Distribution

It is seen now that the design force cycle on the BBSM screw starts out with 25,133 pounds and diminishes to 3,916 pounds. The energy to perform this work in a prescribed time dictates the horsepower selection of the drive motor. To complete the duty cycle analysis one must know the physical performance of the BBSM, the drive motor, and the speed reduction gearbox all of which are selected to the requirements of the duty cycle. In addition, the pushbeam size constraint affected the maximum diameter motor applicable, and since the motor torque diminishes as the diameter is reduced the motor speed rating must increase to produce the required horsepower. This speed increase affects the gearbox ratio required to give the proper lifting time. The procedure was to select a range of lifting times of approximately 15 to 30 seconds, compute the horsepower range and select a BBSM system with the proper properties. This process was iterated several times until a combination of commercial components were found to perform adequately for the feasibility demonstration.

The results of this iteration process has led to the selection of an available small diameter motor which was rated at a nominal 1.5 hp at 10,000 rpm and could produce the required torque at the maximum load condition (appendix B). Normally, a 10,000 rpm input into a gearbox is much too high for efficient operation but since the pushbeam geometry limitation forced this value it was rationalized that due to the short time duty cycle this high speed could be tolerated, especially only to prove the E-MALS concept. The manufacturer's preliminary estimated performance of this motor is shown on figure 8.

Using this motor data and to obtain the lift times, a 40:1 speed reduction gearbox was designed with an estimated overall dynamic efficiency 0.617, thus the design duty cycle was completed and is depicted on figure 9. The values shown depend upon the assumed efficiency of the gearbox (shown on figure 13) which are indeterminant without actual tests. The data of figure 9 also indicates the required motor torque performance at reduced rpm assuming that the gearbox efficiency remains constant. It is seen that it would take approximately 4.6 seconds to lift the plowing plow out of the ground and a range of 6.7 to 11.5 seconds for static lifting, depending upon the plowing attitude. It should be reiterated that the design duty cycle shown on figure 9 was very preliminary and included assumptions, other than the motor or gearbox performance, such as neglecting the inertial loads effects due to acceleration in lifting or any external pushbeam moving friction forces. The actual raising time for a statically loaded plow was determined in the shop during the functional test and are reported in section 5.2. These actual times appear to be approximately 40 percent longer than shown on figure 9.

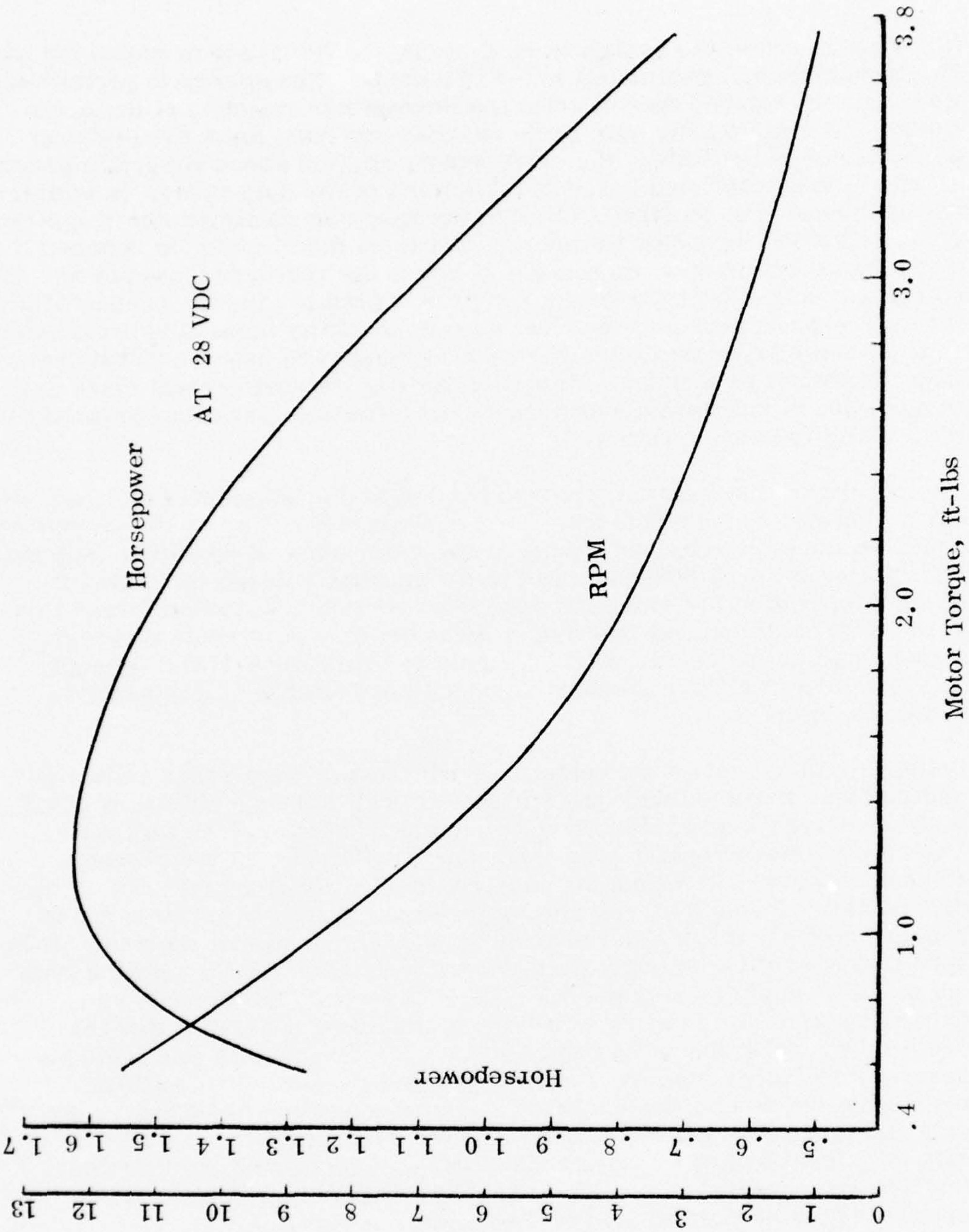

Figure 8. Estimated Motor Performance

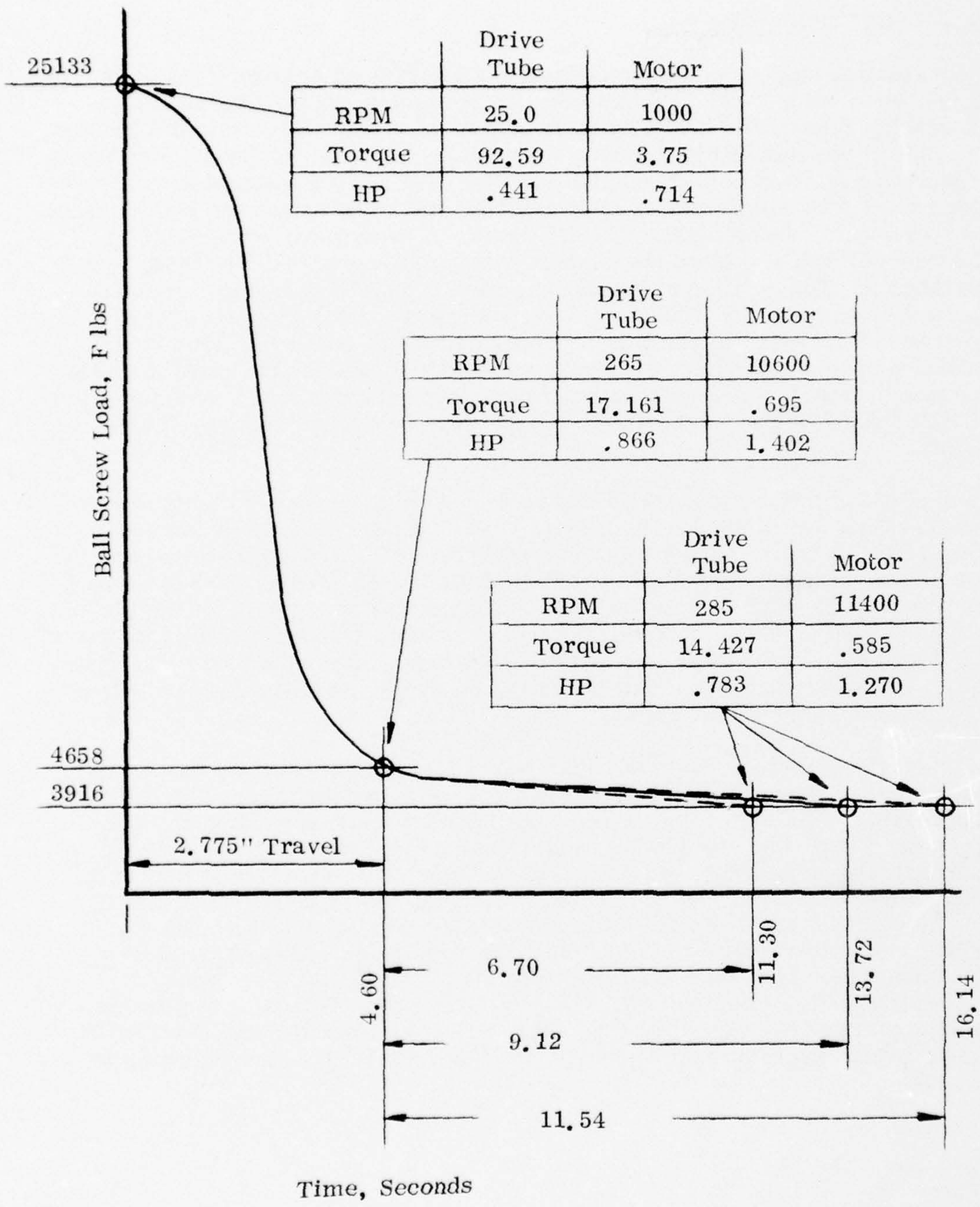

Figure 9. Design Duty Cycle

4.1.3 Mechanical System

The actuator unit is depicted on drawing TD 137761 and figure 1. Besides the mechanical components this unit contains the motor (figure 10), motor reversing relays, the travel position potentiometer and the electrical harness. The nut of the Ball Bearing Screw Mechanism (BBSM) is positioned fixed and receives rotational power from the gearbox output drive shaft tube. The screw translates fore and aft with a 16-inch travel and is pin connected to the present yoke assembly which engages the lift chain. A thrust bearing is provided between the BBSM nut and the support housing to transmit lifting loads to the pushbeam. The position of the BBSM screw is sensed by a small gear set directly coupled to the BBSM nut drive shaft tube. This gear set drives a potentiometer which is electrically connected to the control system. The function of this positioning system is to provide signals to the control system so that the motor speed may be controlled to prevent inertial overshoot of the BBSM nut drive shaft tube, and to provide for sensing up and down travel limits.

A 40:1 speed reduction gearbox (figure 4) is used to provide the proper speed to the BBSM screw motion for lifting the plow in the required times as per the prescribed duty cycle. This gearbox contains a 20:1 reduction worm gear set which provides a sufficient brake effect at the raised plow position to allow the driver time to latch the plow in place. The gearbox along with the positioning gear set also includes a 2:1 spiral bevel gear set necessary to transmit power 90 degrees and a 1:1 spur gear set for geometric positioning of the worm gear set. Two gear shafts with ball bearings, oil seals, and installation hardware complete the gearbox assembly.

The gearbox assembly and the BBSM thrust load bearing housing are integrated onto a heavy base plate channel which provides a structural mounting to the pushbeam. This entire unit is referred to as the housing weldment. This unit was detailed as a weldment to facilitate a short term procurement. A half cylinder cover plate is attached to the housing weldment over the drive shaft zone for mud and debris protection. A thin sheet metal cover plate is used to enclose the gearbox. This cover is indented to accommodate the largest worm possible within the geometry constraint. This cover also contains a pipe plug access for checking the oil level with a dip stick. Approximately 1/3 quart of MIL-C-2105c Grade 80w oil is used to provide a splash lubrication system for the gears. This oil is operational down to -35°F and provides approximately 6,000 hours of use at 125°F before changing is required.

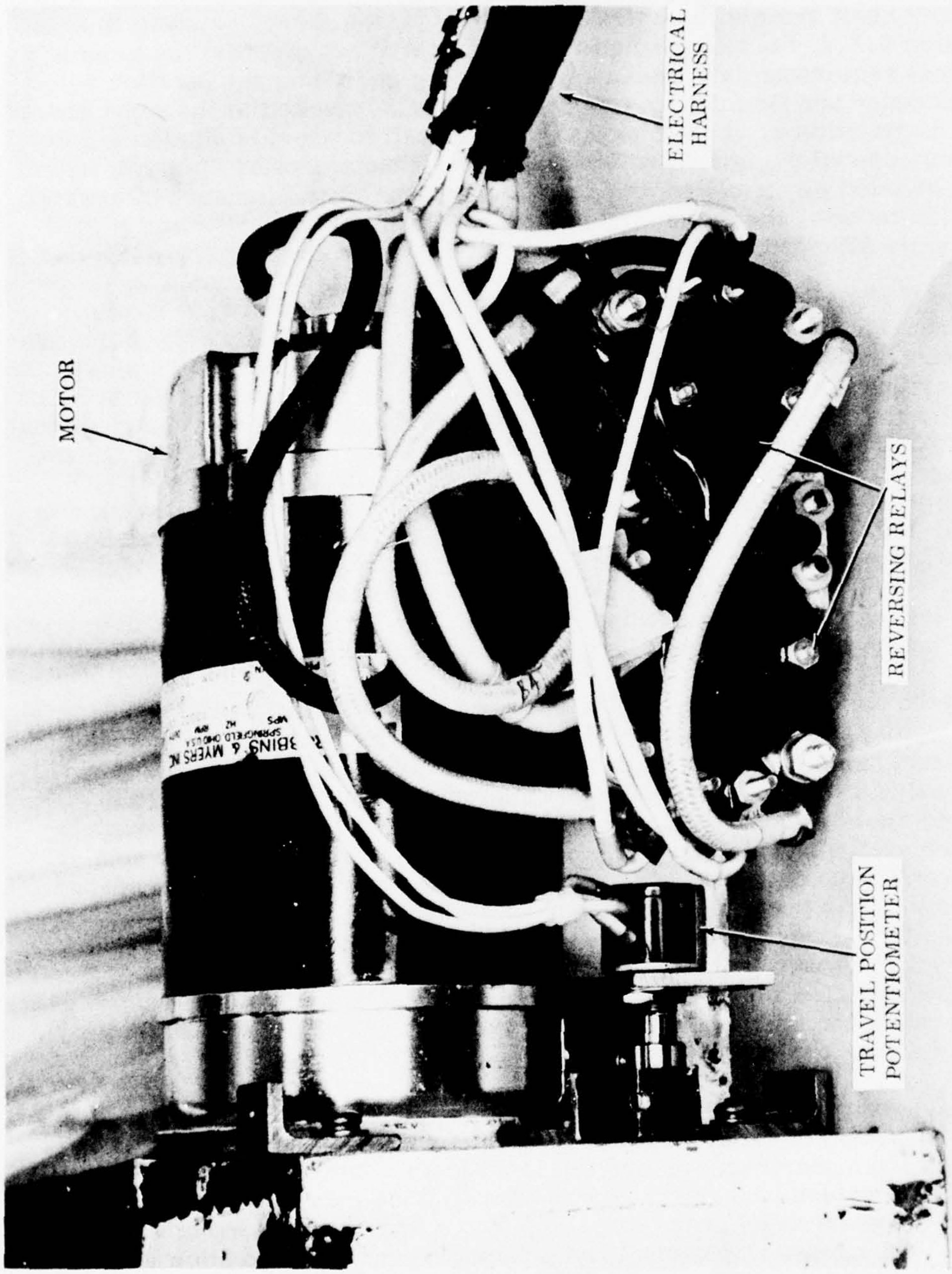

Figure 10. Motor, Reversing Relays, Travel Position Potentiometer & Electrical Harness

A nominal 1.5 hp, 10,000 rpm 28 vdc motor is face mounted to the gearbox and its output shaft mounts the spiral bevel gear pinion. As discussed in detail in section 4.1.4, the motor when functioning will not degrade the vehicle's electrical requirements. The motor reversing relay and the position sensing potentiometer are mounted on a bracket which is attached to the same gearbox face. Angle clips are bolted to the gearbox wall to provide attaching points for a sheet metal mudguard which protects the motor, relay, potentiometer, and associated electrical wiring from the environment. Because of critical space limitations, the mudguard is scalloped at the motor zone and a .020 brass shim filler is bonded in place to match the motor contour.

The BBSM (Appendix C) precision ground screw has a lead of 0.25 inches, a 1 1/2 inch diameter and a 16 inch travel. It has a dynamic load capability of 43 percent above the design dynamic load. Its efficiency is given as a minimum of 90 percent and does not change with load. The rotating bearing nut is bolted to the gearbox output drive shaft assembly. The axial maximum design thrust load of 25,133 pounds is transferred to the pushbeam through a thrust bearing which has a basic dynamic capability (BDC) of 3.94 times the applied BDC based upon 25 rpm and a 20 hour life at the 25,133 pound dynamic load input. A needle bearing is installed at the forward end of the drive shaft primarily as an alignment support point. The maximum radial bearing load at this point is very low (95 pounds) and poses no problem to bearing life. Both the thrust and radial bearings are environmentally protected by shaft seals. The bearings, as well as the ball bearing nut, are grease lubricated via lube fittings.

The gearbox design analysis for the interim test configuration was constrained by two factors; a limited space with a fixed output shaft centerline and the selection of commercial catalog gears and bearings. It became evident early in the analysis that the maximum dynamic torque value of the duty cycle (figure 9) would require catalog rated gear sizes which would not package in the space available. However, since catalog ratings are given for a nominal 8 hour continuous operation and the planned duty cycle is a matter of seconds, this problem was resolved by packaging the maximum size commercial gears possible, then computing the dynamic torque percent overload over the catalog load ratings and then checking with the gear manufacturer as to the gear's overload capability on a time basis. The following analysis depicts this approach and indicates that, in fact, the selected geartrain can take the dynamic torque overload during the short term application indicated by the duty cycle.

The speed reduction of the gearbox was set at 40:1 to produce the proper range of plow lifting times based upon the selected motor performance data. The selected design geartrain was arrived at after a review of possible speed, size, orientation, and load capability combinations. This geartrain consists of a set of 2:1 ratio spiral bevel gears with the pinion mounted on the motor shaft, a set of 1:1 spur gears added purely as a packaging necessity to allow the use of the largest worm gear set possible and which concludes the geartrain. The worm gear is attached directly to the BBSM nut drive shaft tube. The bronze worm gear is only readily available as a right hand gear and since the motor maximum torque is developed during CCW rotation, the ball screw is specified as a left hand screw for raising the plow.

A schematic of the geartrain is shown on figure 11 which indicates the assumed gearset efficiencies. The efficiencies for the spiral and spur gear sets are estimated whereas the worm gear efficiency was computed from catalog data. The overall efficiency is computed to be 0.617. Using these efficiencies and the required values of the duty cycle, table II was computed to show the speed, horsepower, and torque values throughout the geartrain. This data was used along with the catalog rated values to construct figures 12, 13, and 14 which depict the spiral, spur, and worm gearset load requirements vs. the catalog ratings at the rpm speed range. The BBSM screw loads are indicated on each figure. Figures 15 and 16 also indicate a 2.5 times catalog rated capacity. This value was obtained from the manufacturer's engineers who estimated that the catalog ratings, which are based primarily upon the frictional heat generated in a continuous 8-hour operation, could take 2.5 to 3.0 times the torque, or horsepower, for a short time duration of approximately 5 to 10 seconds.

From figure 12 it is seen that the imposed load on the spiral pinion is below the catalog rated curve. Figure 13 shows that there is a 23 percent overload at the maximum dynamic BBSM screw load and then the load falls below the catalog rating shortly after. Figure 14 shows that the worm gear is overloaded throughout the duty cycle, with more overload at the static loading condition. The shape of the rating curve on figure 16 is extrapolated from 1,800 rpm to 5,700 rpm but is considered to be below the 250 percent rating curve since this is the case prior to 1,800 rpm.

Considering that the static load includes an assumed 400 pound dirt load and that the maximum lifting time is approximately 16 seconds (less without the dirt) it is concluded that this overload condition is critical but could function in a controlled test program. In the production design this marginal condition can be avoided if a stronger set of gears is specified.

The maximum dynamic gear loads of table II were used to compute the gear shaft bearing loads which are depicted on schematic figure 17 as resultant radial and thrust loads. The angularity of the resultant radial loads is due to the spur gear tooth force vector which is actually 60 degrees to the horizontal plane. These maximum bearing loads occur, as with the gear loads, for a short time duration. Load values at the static lifting phase can be obtained by dividing the given values by 5.4. Catalog bearings were selected by size constraints and the life rated basic dynamic capability of the bearing at its rpm, load time, and combined effective radial and thrust loads. On this basis, the bearings selected will operate satisfactorily. Bearing no. 3 on the worm shaft is primarily a thrust rated bearing, the other bearings are radial rated bearings which can also take the indicated thrust values.

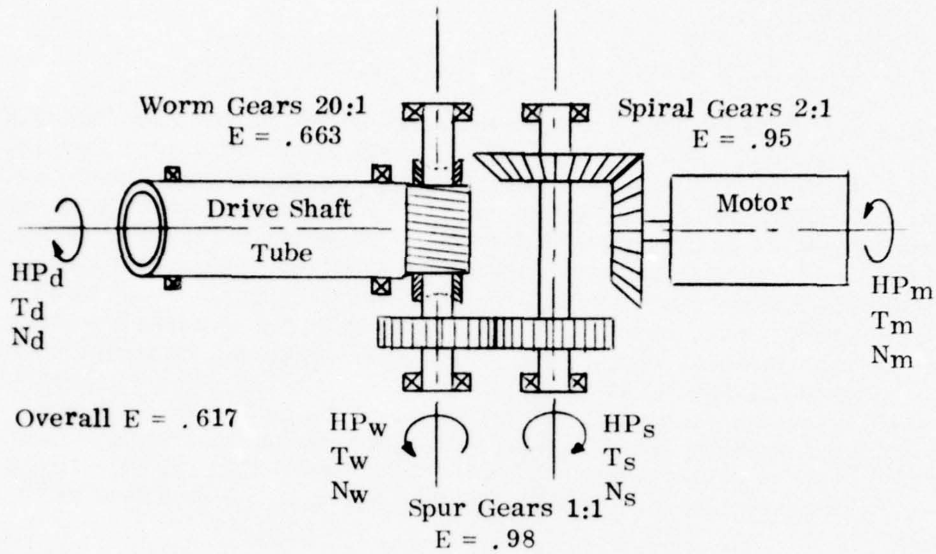

Figure 11. Geartrain Ratios and Efficiencies

Table II. Geartrain Design RPM, HP and Torque (in-lb)

	F = 25133	F = 4658	F = 3916
N_d	25	265	285
N_w	500	5300	5700
N_s	500	5300	5700
N_m	1000	10600	11400
HP_d	.441	.866	.783
HP_w	.665	1.305	1.182
HP_s	.678	1.332	1.207
HP_m	.714	1.402	1.270
T_d	1111.08	205.93	173.12
T_w	83.82	15.65	13.07
T_s	85.46	22.06	13.35
T_m	45.00	8.34	7.02

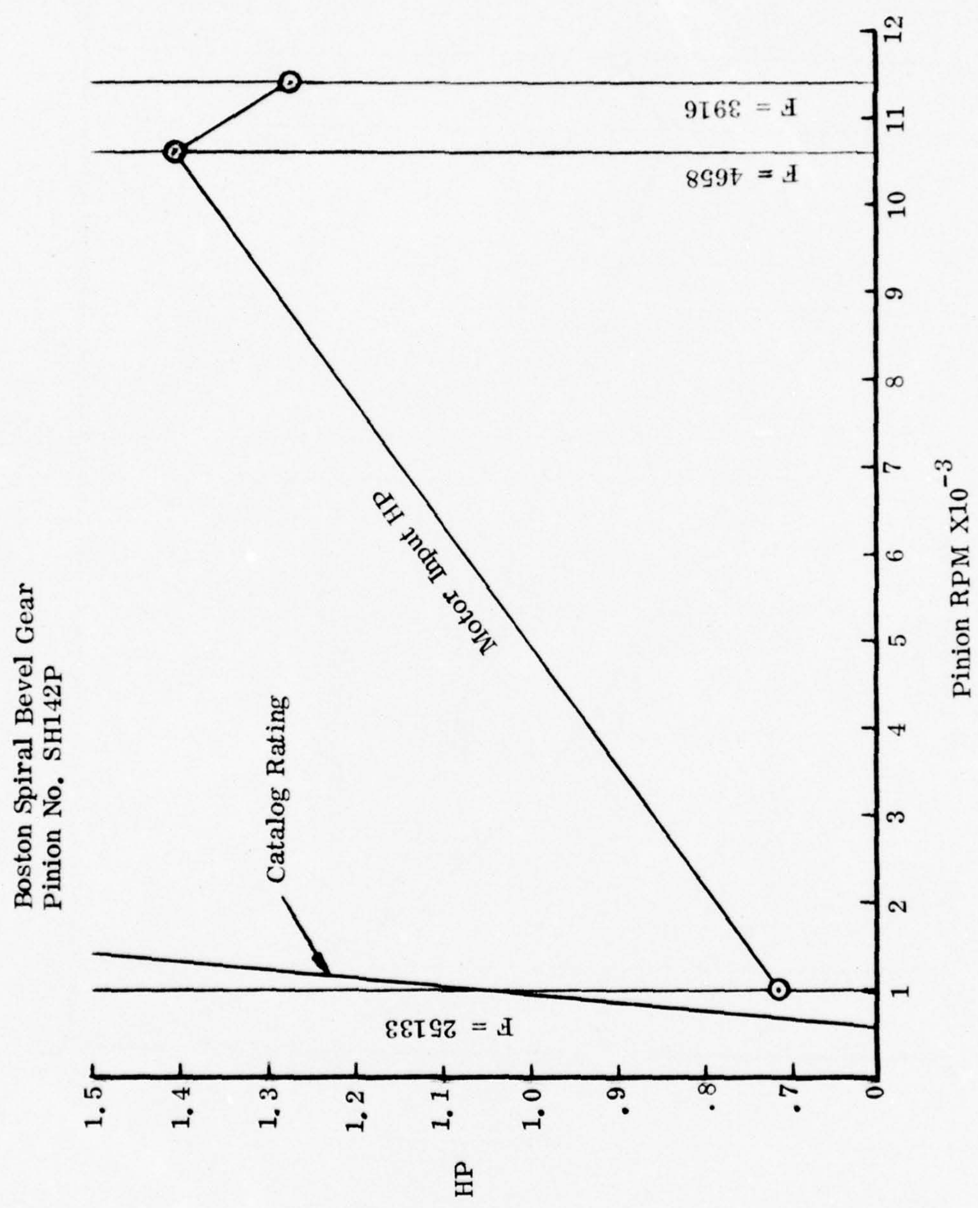

Figure 12. Spiral Bevel Pinion Gear Loads

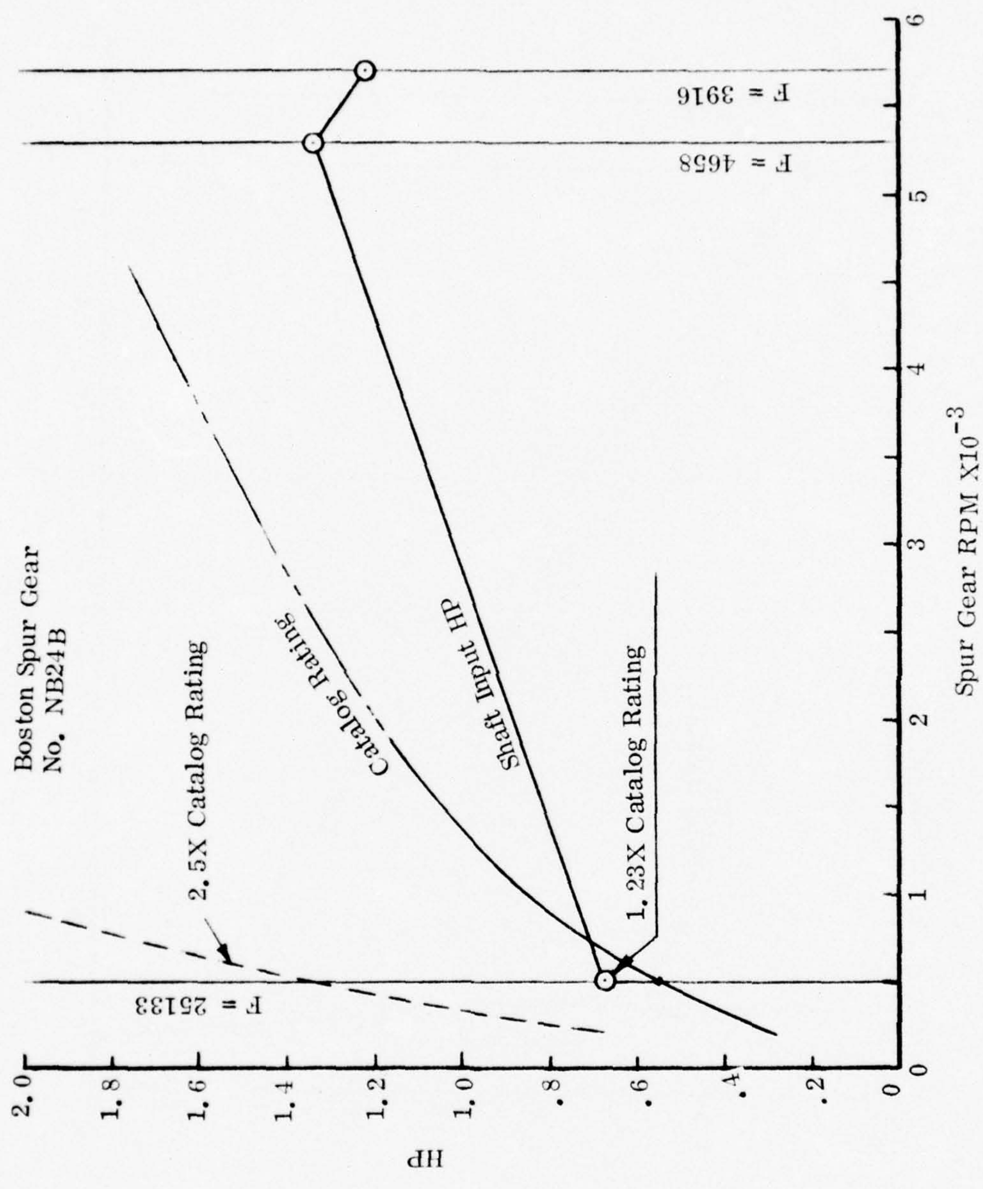

Figure 13. Spur Gear Loads

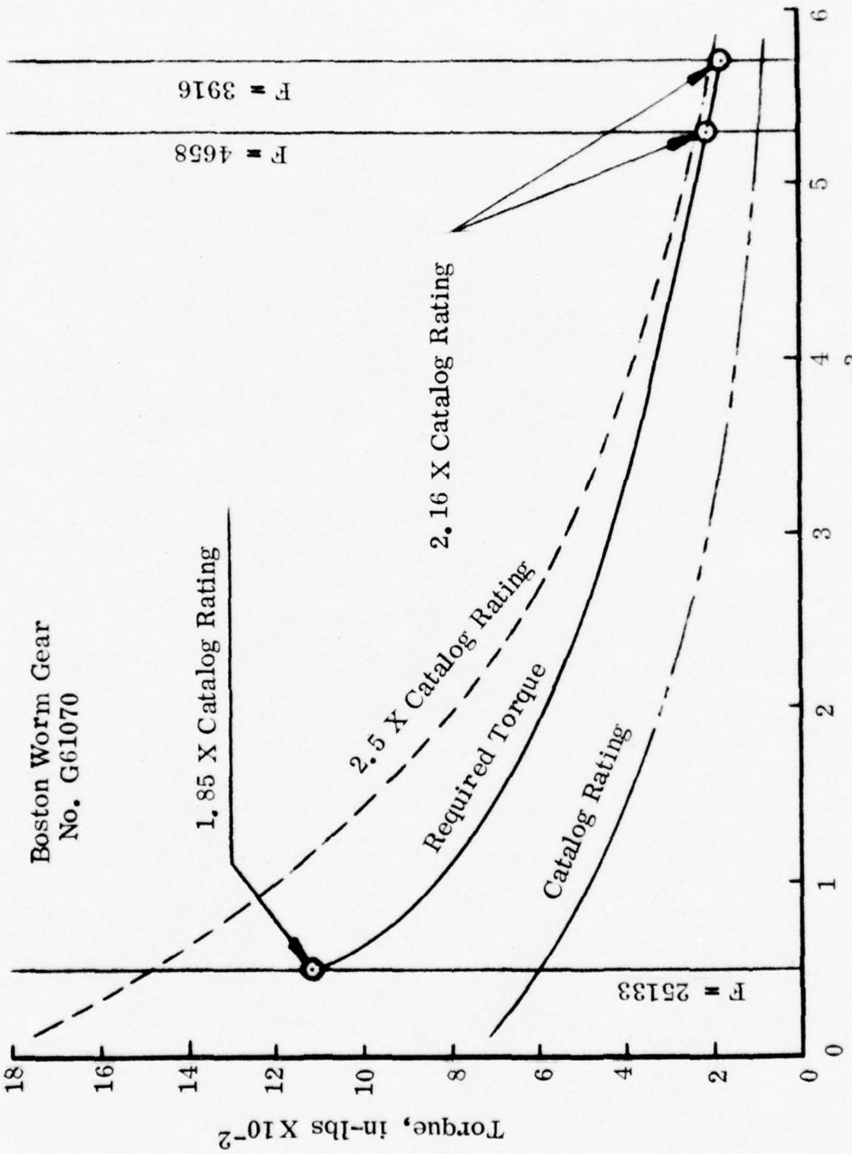

Figure 14. Worm Gear Loads

Pounds Load

		2	3	4	5	6
Radial	55	75	25	113	977	95
Thrust	30	0	854	0	116	0

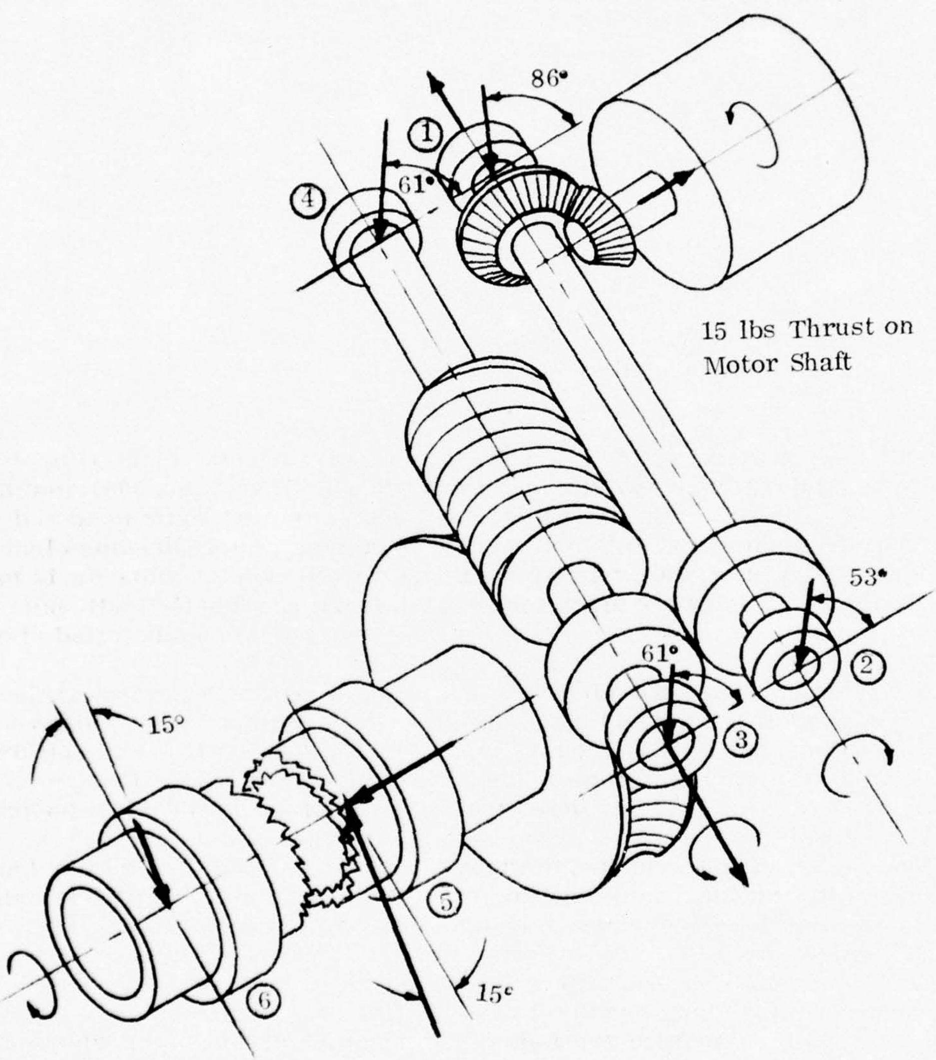

Figure 15. Design Gearbox Bearing Loads

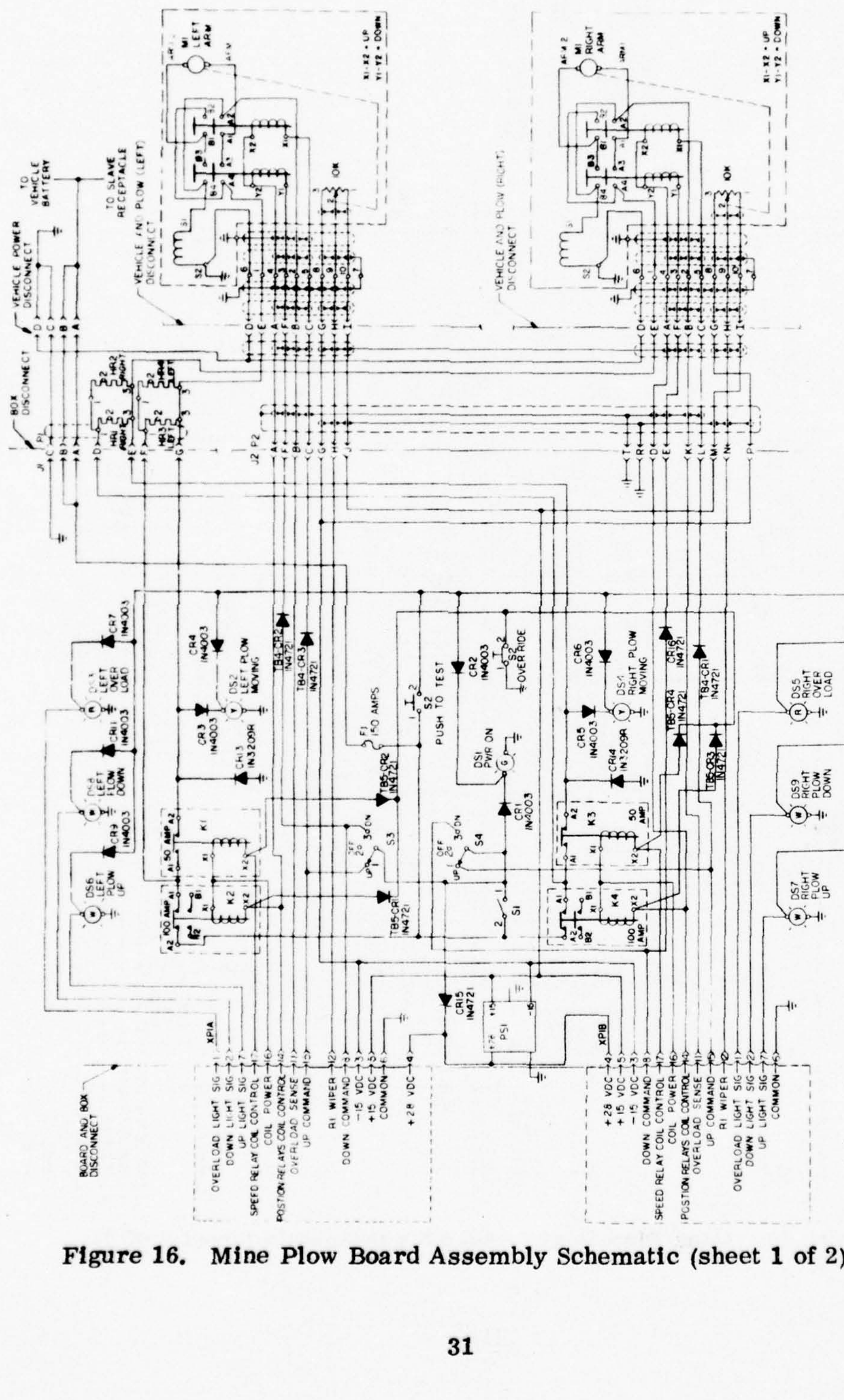

Figure 16. Mine Plow Board Assembly Schematic (sheet 1 of 2)

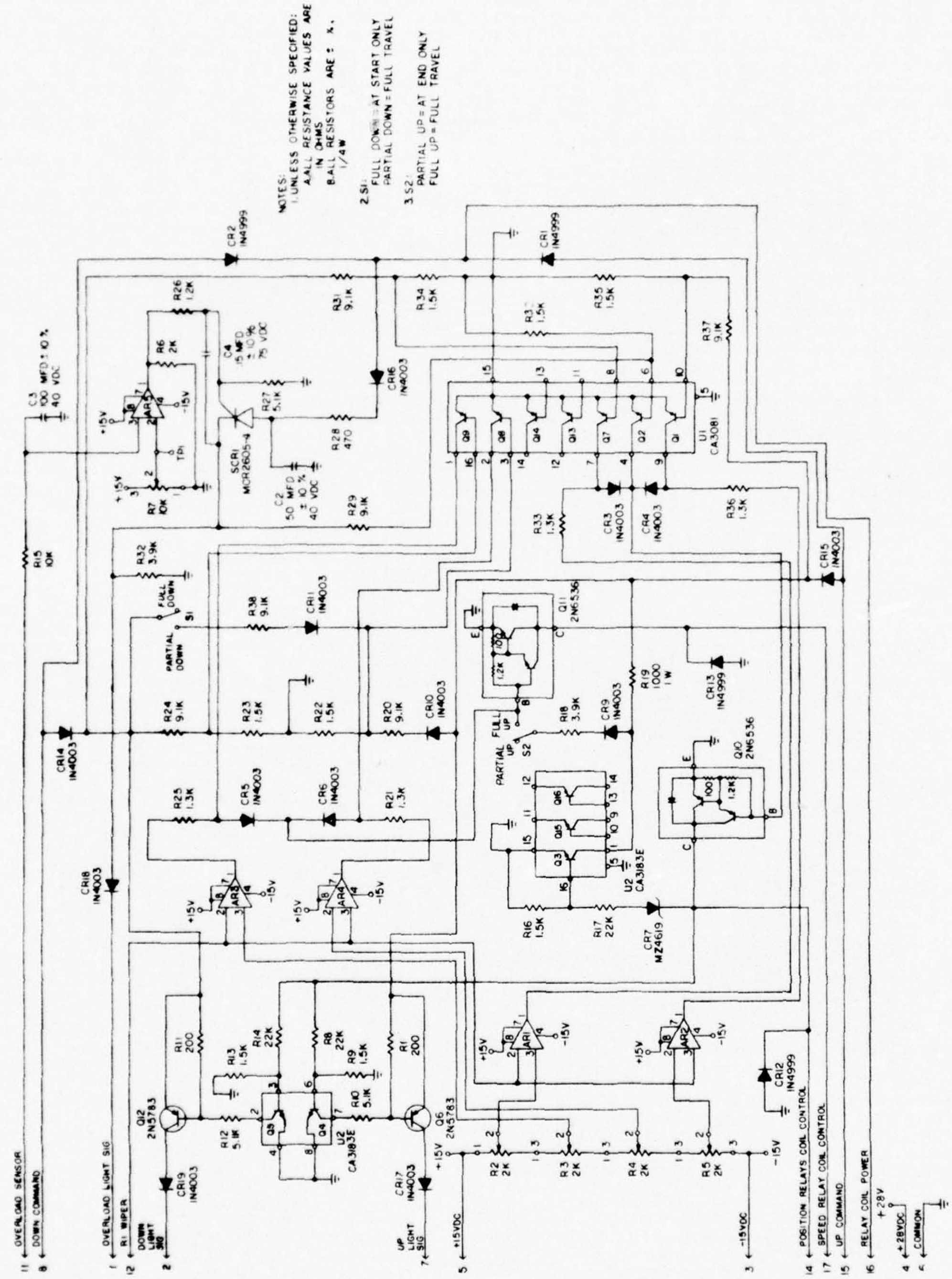

Figure 16. Mine Plow Board Assembly Schematic (sheet 2 of 2)

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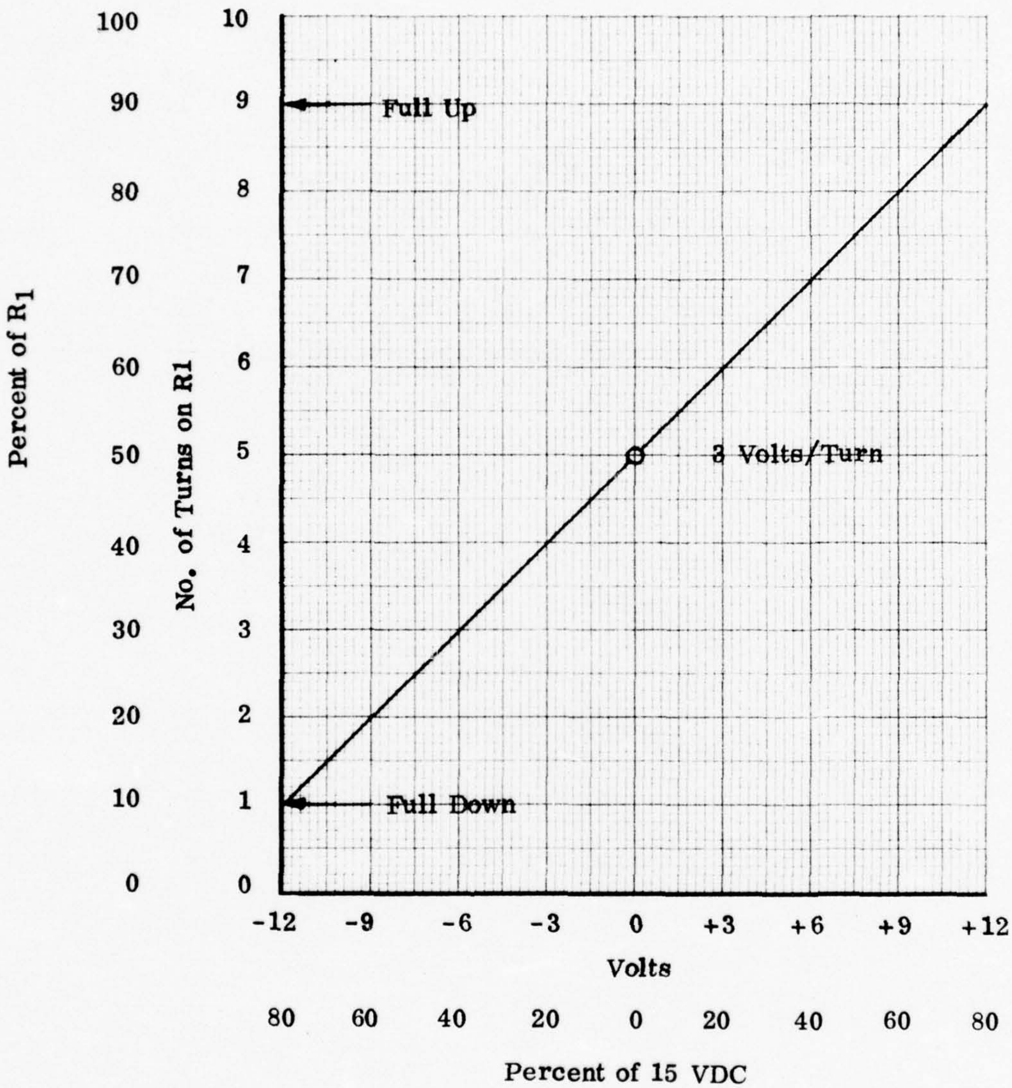

Figure 17. Position Sensor Transfer Function

The bearing shafts are mild steel and were checked for keyway bearing stresses. Standard square keyways are used at the spur gears and the worm gear where the keys can be retained with scalloped keyways. A woodruff key is used at both the large spiral gears. The hardened steel worm is retained by ring retainers.

Conventional gearbox housings are made of cast iron or cast steel to provide good gear shaft alignment and stability. Since our feasibility demonstration test design gearbox does not allow the long term lead time to procure a casting, the design approach was to use a massive wall weldment to be stress relieved before the shaft bearing holes are bored. Vigorous operational usage of this gearbox undoubtedly will cause subsequent distortion of the housing under maximum loads which could cause the analyzed load values on the gears, shafts, and bearings to increase, efficiencies to drop and noise and chatter to appear. Because of this potentiality and the fact that the gearsets could operate in an overload condition, the entire gearbox is considered a critical risk area in the feasibility demonstration test configuration design of the ball bearing screw lifting mechanism.

4.1.4 Control System

The basic control system principle employs analog position sensing (4 positions for each plow) in conjunction with dual speed motor control. The technique is optimum when considering cost, reliability, maintainability, and ease of operation. The system (see the Mine Plow Board Assembly Schematic, Figure 16) provides for independent electrical adjustments of the upper and lower position stops while also providing two intermediate (adjustable) travel points at which motor speed is reduced or increased. An optional adjustment (switch) on each plug-in board permits selection of a full UP and partial DN power mode. These options help provide the flexibility required for the E-MALS feasibility test program. The system provides for setting current overload trip levels at any load values shown on the design duty cycle (Appendix F).

The operator controls on the control panel (figures H1, 2, and 3) perform the following functions:

1. Switch S_2 checks all TWMP indicator lamps.

2. Switch S_1 controls TWMP main power.

3. Indicator DS_1 verifies TWMP main power.

   LEFT OR RIGHT OR BOTH PLOWS ARE SIMULTANEOUSLY CONTROLLED AS FOLLOWS:

4. Switch S_3 or S_4 will provide the plow up position when placed in the up position.

5. Indicator DS_2 or DS_4 will indicate when the plow is moving (motor voltage).

6. Indicator DS_6 or DS_7 will indicate the plow is full up.

7. Indicator DS_3 or DS_5 will indicate if an overload occurs - system is self protecting and no immediate operator response is required.

8. Switch S_3 or S_4 may be left in the up position to automatically re-raise the plow if creep down occurs.

9. Switch S_3 or S_4 may be left in the center position allowing the plow to rest on the lock.

10. The control switch S_3 or S_4 may also be used to manually stop the plow in any partially raised position by placing the switch in the center position when the plow is seen to have reached the desired point.

11. The previous 10 up sequences can be repeated in the down direction by placing S_3 or S_4 control switches in the corresponding down positions.

Since the electrical current required by the E-MALS operation is considerably higher (600 amp peak) than the hydraulically operated system, the power has been taken from the vehicles husky slave start circuit. An additional connector for this purpose is installed and connected to the present slave-start connector. No effect on the slave-start capability will ensue since the E-MALS electrical draw is not required when the vehicle engine is being slave started.

The E-MALS wiring required is run parallel for the most part with existing vehicle wiring thus simplifying the installation. The wiring is brought through the hull at the existing headlamp fixtures. These are modifed, to include a spacer-adapter section with a hole, to permit lead egress and may be removed if the TWMP system is later removed.

The control box installation requires a rearrangement of machinegun, ammo box, driver's instructions, and intercom box on a new mounting bracket. The control box contains wiring, connectors and passive circuits only. The two plug-in cards contain all active circuits, right and left, which are plugged into the control box. The control circuits are identical for each plow and provide all necessary functions in response to operator commands as previously listed in 1 through 11 of the operator controls.

The lifting and lowering prime mover for each plow is a series wound 1.5 hp 28 vdc motor which is energized with two supply voltages. The motor has been designed to operate in either direction, with the design tailored for maximum torque in the lifting mode. This is accomplished by electrical lead. The need for this design approach stems from the packaging problem associated with the size constraints of the pushbeam assembly which houses the motor. The available torque in the reverse direction is reduced due to the electrical lead, but does not result in a system constraint since the required reverse motor torque is less (for lowering plows). Motor direction is controlled by a reversing relay located next to each motor in the actuator unit. Motor speed is controlled by means of a relay and motor ballast circuit which is located behind the control box. The BBSM screw position is sensed by an electrical bridge employing a precision potentiometer driven from the screw drive shaft. The position sensor transfer function and position limits are shown on figure 17. The bridge balance positions (start-stop, slow-fast) are adjusted by our potentiometers on the plug-in cards.

Up and down position commands are initiated by the control switch S_3 or S_4 and by means of electronic control circuits power, the start-stop relays (K_2-K_1* and K_4-K_1*) and slow-fast relays (K_1 and K_3). Normal circuit operation (without use of the mode control circuits on the plug-in boards) provides for slow stops and fast starts. The mode control circuits alternately provide for selection of continuous reduced down power and/or continuous full up power by means of their selector switches. The up-down commands are also used to enable the overload circuit SCR_1 so that the circuit may be reset by placing the command switch momentarily in the off position. The position sensor functions are as follows:

- AR_1 is the "UP" travel position sensor, for limiting the upper travel point.
- AR_2 is the "DN" travel position sensor, limiting the lower travel point.
- AR_3 is the "UP" travel speed sensor, for limiting the upper travel point at full speed.
- AR_4 is the "DN" travel speed sensor for limiting the lower travel point at full speed.

The UP mode selector is a circuit located on each plug-in card with a miniature selector switch S_2. It is necessary to open the control box to change the switch position. The circuit provides the control option of partial UP power at the end of travel or full power for the entire travel. The DN mode selector is also a circuit located on the plug-in card with a selector switch S_1. The circuit provides the option of full DN power at start and during run, ending with partial power or the option of partial power during the entire DN travel.

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*These relays are located in each pushbeam assembly.

For the overload control an electronic clamp is activated by the output of SCR_1 (silicon controlled rectifier) which is primarily the overload indicator lamp amplifier. The SCR will hold an overload alert until the power (command switch) has been removed momentarily. The overload alert is triggered by a comparator which detects motor field currents in excess of any value preset up to the maximum. The input signal to the comparator is the motor field voltage which has been heavily filtered for removal of commutation noise. The trigger signal to SCR_1 is filtered and delayed slightly to prevent nuisance trip-outs from momentary overloads.

The E-MALS system basically operates unbalanced and the UP motor torque required is much larger than the down torque. To provide a means for control of motor and drive train speeds under these variable load conditions the control system also provides for manual preselection of reduced power during rundown. In a similar manner, the max UP power may be manually selected for the entire travel. The power relays then apply full motor power until near end of travel. Near the end of travel the relays turn off reducing power. Whenever the down mode selector switch is set for partial down power, the relays reduce DN power during the entire down cycle. The switch position for partial down power during the full travel appears to be the optimum control mode for use with the E-MALS prior to testing. The switch position for full power to start DN appears to have no immediate need. When the UP mode selector switch is set for partial UP power (at near end of UP travel) normal operation results as previously discussed. The relays then apply full motor power until near the end of UP travel. At near end of UP travel the relays reduce power as previously discussed. Whenever the UP mode selector switch is set for full UP power (for full travel) an auxiliary UP command is given to the electronic controls for the entire UP cycle. This auxiliary command is derived in part from the main UP command signal and it will maintain full UP power until end of travel. At travel end, it is switched off. The switch position for partial power at end of UP travel (normal mode) appears to be the optimum control mode for use with the TWMP prior to testing. The switch position for full UP power during entire travel might find use to raise the plow into the lock mechanism if reduced motor power from the ballast resistor is found too small when adequate for DN travel.

The overload control and indication in case of a stuck or jammed plow is provided by the turn-off of the power relay coils. This action is automatic and prevents burnout of the motor from overcurrents. Other circuit faults such as shorted cables from an exploding mine, for example, are protected by an internal fuse which disables the entire system. The current overload condition is indicated by an indicator lamp on the control panel. When triggered the indicator stays on until the power source (up or down command) is momentarily turned off. The circuits also include a time delay which prevents drop-outs from noise or transient overloads and provides for stable control. The overcurrent signal is derived from the output of a voltage comparator which may be adjusted by an internal adjustment for any motor current. The voltage drop of the motor field is thus used to trip the overcurrent gate when exceeding the

settings. The settings are made using the curves of figure G-1 on page G-4 to obtain the desired maximum motor current as read from the curve. This value of motor current will then trip the overload circuit whenever the overload period (current) exceeds the time delay built into the circuit to prevent nuisance trip outs. The corresponding motor torque may be read from figure B-2 on page B-5. The motor field voltage signal to the over-current gate is heavily filtered because of the motor commutation noise.

System performance indicators also include a plow moving indicator and plow up and down position indicators The plow moving indicator will light brightly when full voltage (full torque) is applied to the motor circuit and will light dim when the power reduction relay operates. The plow UP and DN indicator circuits can operate only when either the UP or DN control switch is actuated. The full UP and DN positions are indicated by lamps with their control circuits powered from the UP and DN commands. The power is applied to the lamps by the control circuits when the end of travel is reached which is indicated by the position sensor circuits.

4.2 LIFT CHAIN AND BOOT SYSTEM

The DT-II EPR's on lift chain failures reveal three problem areas: The chain cannot take lateral flexual loads, the bare carbon steel chain rusts, and the chain leaf link interstices act as carriers to bring mud and debris into the pushbeam cavity. The solution to the lateral load problem was solved by using a pin joint at the upper chain attachment point. The solutions to the rust and debris problem required a survey and evaluation of various concepts and approaches.

An initial chain antirusting approach tried was to clean and degrease, apply a phosphate coat per MIL-P-16232 and finish with a baked-on-solid film lubricant per MIL-L-46010. This was done at Ft. Knox on completely assembled chains. Unfortunately, this process could not protect the metal surfaces at the chain components which touch each other, thus rusting still occurred at these points.

Several approaches were analyzed to find the most expedient and least costly method to prevent the rusting and debris problem. Some of these included use of an all stainless steel chain, design of a more effective chain mud cover, filling the present chain voids with an elastomer, installation of automatic oilers on the pushbeam, use of reinforced plastic belting, use of thin stainless steel strap belting, use of braided flat belting, use of a cable rope array, protection of the chain with a bellows or boot, and various combinations of these. Most of these approaches were discarded as being too costly, impractical, high design risk, or too long of a lead time to apply to the scheduled field test program of the E-MALS system.

The best approach under the circumstances was to select the protective boot system (figures 18 and 19). This approach presumes that if the present chain, originally greased, is protected from the environment it should not readily rust and the boot itself will prevent mud and debris from entering the pushbeam. Along with this presumption, the pushbeam itself would be reworked in a manner to hinder any mud or debris from entering from other directions.

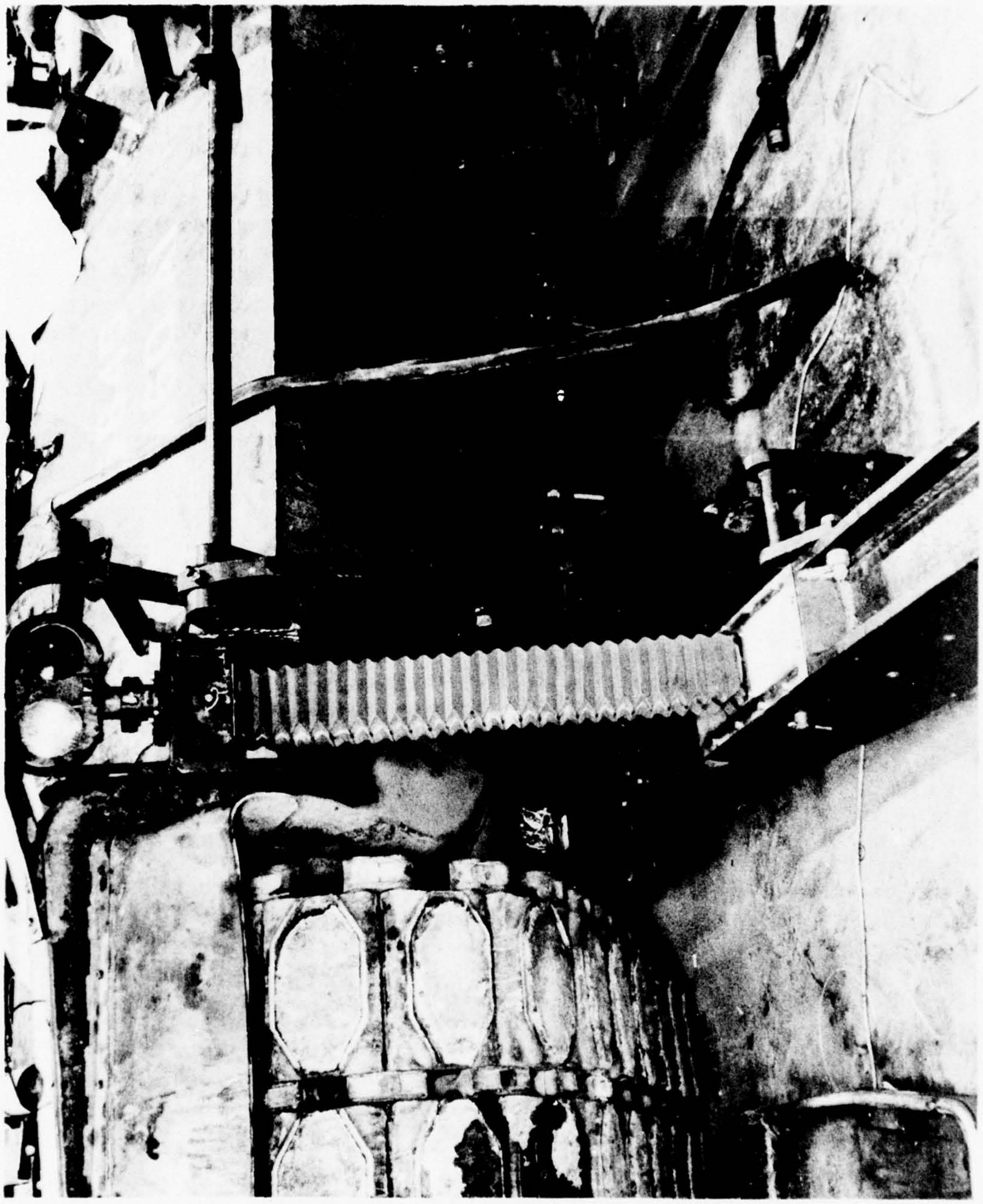

Figure 18. Protective Boot System (Lowered)

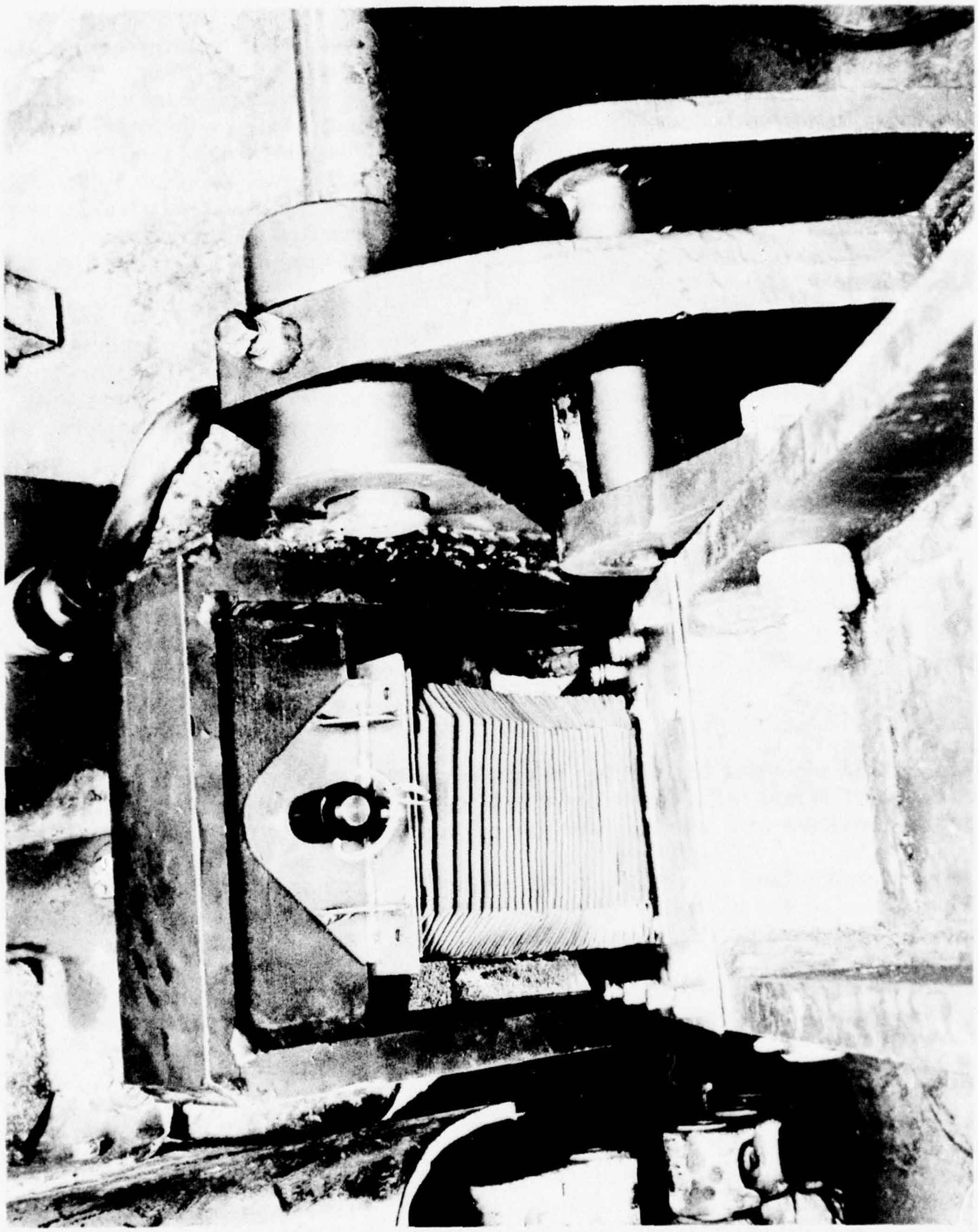

Figure 19. Protective Boot System (Raised)

The boot concept had one major drawback. The boot, when compressed, requires a finite space and this space was not available on the TWMP lift system in the plow up position. Either the upper chain attach point had to be extended to accommodate this compressed boot dimension, or the maximum plow-up position had to be lowered and a longer travel lock hook made to reach the lower positioned plow. The first alternative was selected, primarily so as not to degrade the vehicle's stowed plow mobility. Accordingly, the upper chain mounting point was raised on the mounting bracket with an appropriate weld-on bracket. A quick disconnect pin is used at this upper joint to eliminate the side flexural load on the chain.

The lower end of the boot is attached to a sliding plastic plate to accommodate the change in chain orientation as the pushbeam is raised. This plastic plate moves in a channel bracket which is bolted to the pushbeam. The entire boot assembly can be rapidly replaced in the field by removing this channel bracket and the upper chain pin connection.

4.3 MOUNTING BRACKET

The total effort extended on the TWMP mounting bracket considered three separate applications: for the M60A1 tank, the M728 CEV, and a quick disconnect mounting bracket application which was actually layed out for both the M60A1 and the M728. The quick disconnect concept was eliminated early in the program due to funding limitations and will not be discussed herein.

4.3.1 M60A1 Vehicle

The M60A1 mounting bracket has been changed by adding structural material to correct DT II test failures, to provide a path for the E-MALS electrical harness and to provide a new mounting design and location for the lift chain.

The structural beef-up was accomplished at Ft. Knox and these changes were documented on the mounting bracket drawing as Revision "A." A NASTRAN model of the Revision "A" mounting bracket was made (Appendix E), as a check of the structural fixes. The modeling has indicated that, based upon the assumed loads analysis, case 1 in appendix D, positive margins occur throughout the structure of revision A. The drawing changes brought about by the incorporation of the E-MALS and chain/boot systems are documented as revision B.

As an expedient measure, the E-MALS electrical harness was designed to snake through the mounting bracket which would act as protective conduit. This approach necessitated 1-inch diameter holes be burnt through both inner diagonal structural webs. In addition, the harness had to pass through the 4-inch tube and the pushbeam connector. This necessitated a 1-inch slot be cut in both these components. A stress analysis of these hole areas has indicated that the structural integrity of the mounting bracket is still maintained.

Future product improvements of the E-MALS system should consider using a separate harness conduit not only to eliminate the holes in the mounting bracket structure but to provide a simpler installation and maintenance of the harness assembly which presently terminates at a difficult access location of the terminal block on the mounting bracket. Incidentally, this terminal block caused interference with the connector bolts during the in-house functional test program. As a quick fix, the bolt heads were ground down to a 1/8-inch thickness. This was structurally acceptable since the bolts primarily act in shear under operational loads.

An arch type bracket was welded to the mounting bracket to provide a structure to mount the extended lift chain/boot system. This bracket allows the lift chain and boot to be pin connected. The add-on bracket was stressed for proper sizing. The stress analysis approach was to make the quick disconnect attach pin the weak link in the attachment. Future product improvement of the mounting bracket should consider an integral structure to eliminate this weldment chain bracket.

4.3.2 M728 Vehicle

The M728 mounting bracket has been changed by adding structural material to correct DT-II test failures and to accommodate a new pin connected chain/boot attachment similar to that done on the M60A1 mounting bracket. The structural modifications were based upon the results of the NASTRAN analysis for the case 1 load application (appendix D and E).

Since the 4-inch tube on the mounting bracket is located too far from the hull for an effective center clevis installation, the tube bending failures were solved by increasing its section modulus. A tee section is welded along its length to provide additional bending resistance in the direction of loading.

Distortion failures of the vertical sides of the mounting brackets are eliminated by the addition of a structural plate along its backside, which essentially results in a channel section. This addition is deemed sufficient to carry the case 1 loads imposed.

The upper end of the structure is extended into an arch for the required chain boot clearance. A quick disconnect pin joint for the chain is provided.

4.4 PUSHBEAM

The changes to the pushbeam include structural modifications to correct DT-II test failures, changes to accommodate the E-MALS actuator unit, and alterations to install the chain/boot attachment bracket. The basic shape and size of the pushbeam was not altered.

The structural modifications were accomplished at Ft. Knox and those changes were subsequently documented on the pushbeam drawing as revision "A." A NASTRAN model of the revision A pushbeam was made, appendix E, as a check of the structural fixes assumed as necessary. The modeling has indicated that, based upon the assumed loads analysis, case 1 in appendix D, positive margins occur throughout the structure of revision A. The drawing changes brought about by the incorporation of the E-MALS system and the chain/boot system are documented as revision B.

The new actuator position also necessitated a relocation of the lift chain attachment point, for access, and this had required that a longer chain (16 more links) be procured. The existing unused attachment holes in the pushbeam were plugged with dead bolts as an aid in keeping out debris.

Since the ball screw actuator moves linearly by a screw action driven by the input torque of the drive nut, the reaction torque on the screw/roller had to be eliminated in order to maintain the proper attitude of the lift chain. Antitorque hardened steel rails are installed on the inner vertical walls of the pushbeam along which the roller pin rides. The roller pins have been lengthened to accommodate this approach. Both the pins and the rails were liberally greased for the in-house functional test program. The reaction point load on the rail was computed to be 51 pounds when the actuator produced its maximum rated torque.

The boot installation required an extensive rework of the chain entrance area of the pushbeam. This zone was made a level section to accommodate a flat channel adapter to carry the sliding member of the boot (as discussed in section 4.2). The mud cover assembly on the small roller was discarded but the roller itself was used as is. The structural integrity of the pushbeam at the boot attachment zone was not affected.

During the in-house functional tests it was observed that certain conditions can exist wherein the lift chain is limp and could fold down to lay on the ball screw drive nut. When the actuator is actuated and before the chain slack could be taken up the chain would rub against the nut's ball return races and a possibility existed that jamming could also occur. As described in section 5.1, an expedient metal guard was fabricated to prevent this possibility. The pushbeam assembly drawing depicts a similar type of guard but attached in a more positive manner. Any future product improvement program should consider a design assembly wherein this separate guard is not necessary.

4.5 CENTER CLEVIS

During DT-II testing certain mounting bracket failures occurred when the 4-inch diameter tube bent in the middle. A field modification at Ft. Knox to correct this deficiency was a support structure attached to the vehicle's hull to stabilize

the 4-inch tube. This add-on unit was called the center clevis and consisted of an adjustable yoke and tongue arrangement which straddled the tube and was assembled with a quick release pin.

The center clevis concept functioned correctly to prevent tube failures but the unit itself created some undesirable shortcomings, notably the long quick disconnect pin had a tendency to take a permanent bend and made disassembly/ assembly very difficult. In addition, the unit was very heavy and the adjustment bolt was difficult to work when it became rusty.

To overcome these shortcomings, CDD instituted a preliminary design study to arrive at a more efficient manner in which to prevent bending of the mounting bracket tube. The study included an input design concept from MERADCOM and ranged from simply increasing the tube's section modulus to using a section of extra lift chain as a strap hold on the tube, and included various alternatives of the present clevis concept, primarily addressed to the adjustment problem. These studies were based on using the case 1 plowing loads (appendix D) from the NASTRAN program produced the forces shown on figure 20. The final approach selected was based upon cost and expediency to meet the scheduled field testing of the E-MALS TWMP.

The new center clevis design is very similar to the original design. The unit size and weight have been reduced, the quick release pin was shortened considerably, and the adjustment bolt, with a new jam nut, was arranged for access with a box wrench. This unit was fabricated and shipped to APG along with the TWMP test system.

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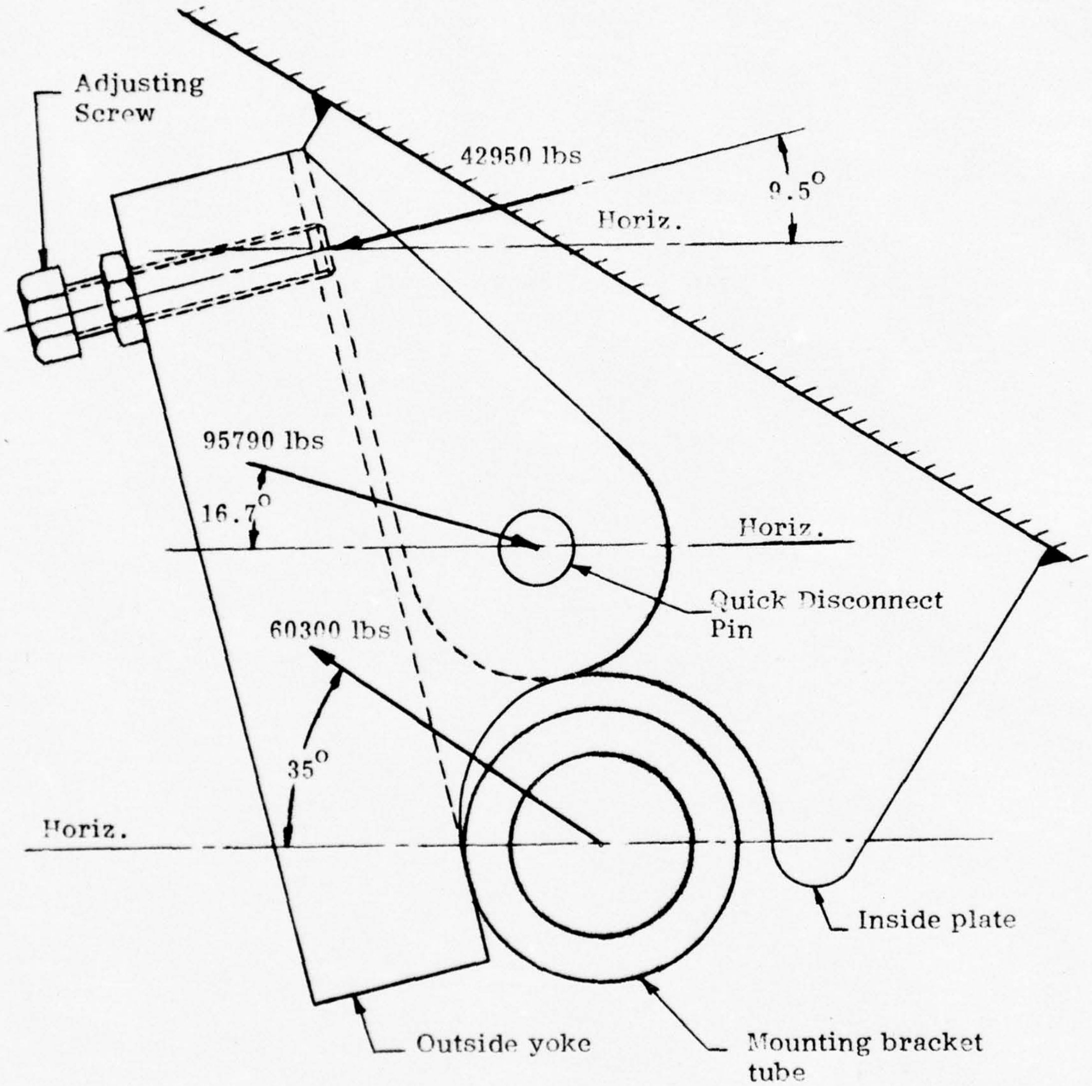

Figure 20. Center Clevis NASTRAN Design Forces

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5.0 RESULTS

The E-MALS drawing package was completed in an expeditious manner and hardware procured to assemble two concept demonstration units. The scheduled in-house plan in preparation for delivery of one complete TWMP system for field testing prior to the DEVA IPR was to install the TWMP system on an available M60A1 tank, check and preset the E-MALS system, and finally to strip the TWMP down as little as possible for shipment to APG for immediate installation on the field test vehicle. The objective of this plan was to reveal any unforeseen design, assembly, checkout, and functional discrepancies and to correct them.

5.1 FABRICATION, ASSEMBLY AND CHECKOUT

A design discrepancy which showed up during assembly of the actuator gearbox was that the output shaft turned in the wrong direction for the lifting function coupled with the worm shaft thrust load bearing at the wrong end of the shaft. Three possible solutions to this discrepancy were to, 1) use a left hand BBSM, which was eliminated due to the long lead procurement time, 2) change the motor's high torque output to the CW direction, which was also eliminated due to the long lead time (and cost) to rewind the armatures, and 3) to change to a left hand worm gear set. This last approach was selected as the least costly and quickest solution, but not without penalties. The left hand gear set replica of the right hand gear set only was immediately available with a cast iron worm gear which is approximately 14 percent less efficient than the bronze right hand worm gear. The use of this left hand worm gear set also necessitated the addition of a thrust bearing assembly on the worm gear shaft. The gear box as originally designed can be made to function correctly if only the BBSM is specified as a left hand thread. This will maintain the original motor CCW high torque direction and the worm shaft thrust load into the original thrust bearing.

A design oversight was revealed when during functional checkout it was noticed that a limp lift chain could jam into the rotating BBSM nut. A quickly conceived sheet metal guard was installed over the BBSM nut as a protective device using existing available holes in the pushbeam as a mounting point.

The checkout procedure of the E-MALS system consisted primarily in setting the travel limits of the actuator prior to installation into the pushbeam (appendix G). A bench setup was constructed for this purpose. The position potentiometer has a capacity of ten physical turns, eight of which are used to measure the full 16-inch travel of the actuator. The procedure was to correlate the null point of the potentiometer (5 turn point) with the center of the actuator travel. Travel limits are then set with the potentiometers on the control circuit card: The DN travel point about 1.0 inches of actuator screw travel before hitting the screw nut and about 16 inches of full screw exposure for the initial UP limit. The final UP limit is adjusted on the vehicle to allow proper latching of the raised plow. Since the demonstration model BBSM has no physical stop for extended travel, the initial 16-inch setting is to prevent overextending the screw with loss of its ball bearings. Future BBSM units will have a specified physical travel stop.

5.2 FUNCTIONAL AND QUALIFICATION TESTS

The primary objectives of the in-house functional and qualification tests of the E-MALS system were to reset the plow travel limit potentiometers, set the travel speed potentiometers and ballast load, and exercise both plows in a prescribed manner. In actuality the first two objectives were performed during the plow exercising sequences since the travel limits and speed had to be readjusted as the applied load on the plow changed.

The qualifying exercise test was specified as ten cycles at 20 percent of the maximum design duty cycle load (25,133 pounds) and ten cycles at 35 percent of the maximum design duty cycle load. During the cycle, the plow was to be raised just far enough to prove observable steady state operation through the horizontal position of the pushbeam. During the lowering portion of the cycle the load was to be relieved to 18 percent, which represented the maximum static load (400 lbs. dirt) of the duty cycle.

At the initial installation of the TWMP, both plows were exercised frequently, first with pushbeam only, then with the moldboard and skid shoes attached, to burn-in the gear system and to check the operation and clearances. As a result of these initial exercises, it was found necessary to grind out a considerable portion of a weld on the mounting bracket to clear the required chain boot mounting flange at the raised position. Also, it was observed that a guard was required to protect the BBSM nut from being jammed by the lift chain. Most importantly, it was observed that the actuator motor case temperature, which was instrumented, rose quite rapidly at each operation. This temperature rise paced the functional test program.

The initial exercises were halted when erratic behavior of the travel limit stops were observed on the left hand unit (called #1, the right hand unit was called #2). The actuator was removed and examined. The problem was caused by the worm gear square key which had worked itself out and had jammed the travel position gearset, which in turn had destroyed the travel sensing potentiometer. Both actuator gearbox keyways were reworked for positive key retention, new travel position gearsets and a new potentiometer for #1. (NOTE: The keyway descriptions given in section 4 reflect the correct design as a result of this failure.) The actuators were reinstalled in the TWMP system and the burn-in exercises continued in preparation for the qualifying load tests.

To simulate the 20 percent load condition a 475 pound weight was placed on the plow with a 54-inch moment arm from the pushbeam hinge point. This load was left on the plow during the lowering phase. At a 35 percent condition, 1,690 pounds was added at a 55-inch moment arm for the raising phase and all weight but 300 pounds was removed prior to lowering. During these qualifying cycles, the motor case temperature and later the motor amps and voltage inputs were measured for the UP cycles only. The following paragraphs describe the chronological events and results of the qualifying functional test program:

- 20 Percent Load Test on #1. The complete plow was exercised up and down a few times to check and adjust the travel limit cutoff points. The 20 percent load of 475 pounds at 54 inches was laid on top of the moldboard and the plow was run up and down ten consecutive times to an average height of 9 inches off the floor (as measured at the outboard tooth of the moldboard). The motor case temperature was 85°F at the start and 101°F at the end of the ten cycles. It reached a maximum of 107°F before it began to cool down. The motor voltage and current were not recorded. The test then proceeded to the 35 percent load.

- 35 Percent Load Test on #1. A total weight of 1,690 pounds was placed on the moldboard at a 55-inch arm. The plow was run with this weight in the up direction to an average height of 4 to 6 inches off the floor, stopped, and 1,390 pounds removed and ten run down with 300 pounds remaining. Ten cycles of this routine were run consecutively. The motor case temperature at the start was 97°F and 110°F at the end of the 10th cycle. It reached a maximum of 130°F before cooling down. The motor voltage and current were not recorded. The #1 no-load raising time was checked with a wristwatch at 14 seconds to travel 22 1/2 inches of chain which gives an average rate of 1.6 in/sec.

- 20 Percent Load Test on #2. The complete plow was exercised up and down a few times to check and adjust the travel limit cutoff points. A total weight of 475 pounds at 54 inches was laid on top of the moldboard. The motor case temperature was noted to be 111°F. Two up and down cycles at an average height of 10 inches off the floor were run when it was observed that the motor case temperature was rising rapidly and the motor sounded like it was laboring. The test was stopped for evaluation of this condition. No motor voltage or current were recorded during these first two cycles. It was decided that the gearbox was too tight and required a burn-in phase. The load was removed and the plow was exercised full up and down four to five times and the burn-in was stopped when the motor case temperature rose to 181°F. The actuator was removed from the plow, disassembled and examined. The motor had failed due to one brush wire clamp rubbing on the commutator. It was postulated that this failure may have been caused by forcing the brush holder down during assembly of the motor cover or by the burn-in

exercises which applied shock loads to the plow when the limp chain slack was taken up. The motor was sent back to the manufacturer for rework. (NOTE: It was decided that at this time the motor voltage and current were to be monitored for those conditions which were felt to be pertinent.)

- After motor #2 was received back from the manufacturer it was installed on the actuator. The actuator was bench checked and installed into the #2 bushbeam only (no moldboard and skid shoe). The unit was exercised up and down several times to burn-in the gearbox. The initial motor case temperature was 79°F and rose approximately 10 degrees during this burn-in cycling. The motor "up" voltage was measured at 25 volts and the current was 20 amps. The moldboard and skid shoe were attached to complete the #2 plow and a new burn-in cycle was started. One cycle was run with an initial temperature 98°F and a final temperature of 103°F. During the up phase, 24 volts and 62 amps were recorded. It was decided that the 20 percent load test on #2 should proceed. The 475 pound weight was loaded at the 54-inch arm. The initial motor case temperature was 103°F. Ten consecutive cycles were run to a height of 15-inches off the floor with a momentary pause at cycle number 6 when the overlight came on. The temperature at this point was 119°F and 45-55 amps were observed. No voltage was recorded. The reason for the overload light was not immediately determined. Run No. 7 was started at 127°F and after completion of the tenth cycle the temperature was 141°F which then rose to a maximum of 155°F. The amps were constant at 45 to 55 throughout the ten cycles. No voltages were recorded.

- 35 Percent Load Test on #2. The motor was allowed to cool down. A load of 1,690 pounds at a 55-inch arm was placed on top of the plow. The plan was to raise the plow approximately six inches, stop it, then remove all but 400 pounds for the down phase. The initial motor case temperature was 88°F. Two cycles were run and the test stopped when noticeable motor laboring was heard together with a low frequency oscillatory noise from the actuator. The motor case temperature at this time was 118°F. It was decided to again burn-in the gearbox. The 35 percent load was removed and the plow was exercised full up and down for 12 cycles. The maximum motor case temperature reached was 199°F and testing was stopped for the day. It was postulated at this time that the oscillatory noise was out to an adverse stack-up of tolerances on the actuator drive shaft/ball screw flange/gearbox housing interfaces and the motor labored perhaps because its performance was less than the #1 motor. The 35 percent load test was resumed at cycle #3 on the following morning. The initial motor case temperature was 77°F. At the first cycle (#3) the overload lamp lit on the up phase when the plow

reached 1 1/2-inches off the floor. The test was stopped to reset the overload point on the control board. The test was resumed with cycle #4. The motor case temperature was 86°F and the 10 cycles were completed at a final temperature of 116°F. In each cycle the overload light again came on at 1 1/2-inches off the floor (which stops the motion) and the plow was then lowered. The motor measured 21 volts/95-105 amps for these cycles. The reason for the overload light to come on at the same plow position was not immediately determined but it was suspected that some rubbing resistance of the pushbeam hinge point may have been the cause. It was observed that this plow twisted more than #1 upon raising. The plow hinge points were liberally oiled for future tests.

- The 35 percent load was shifted to #1 actuator for a one cycle check. The plow was raised 6-inches off the floor and 22.5 volts/65 amps were recorded.

- Timing tests were run on both units at no load. Table III gives the results. The total travel distance is approximately 23-inches, but the times recorded were from the start switch actuation, which included some slack chain take-up time.

At this point, it was decided to test at a higher capability of the E-MALS since the tentative field test plan had indicated operations at a 50 percent load setting.

- A 2,560 pound weight at 49-inch moment arm was set on the #1 plow. This value represented 42-1/2 percent of the maximum design load. The plow was raised a nominal 4-inches with 22 volts/58 amps recorded with a cold motor.

- Prior to further testing on #2, the ball screw attachment to the gearbox drive nut was rotated 120° in an attempt to eliminate any possible tolerance stackups which may have caused the oscillatory noise. In the tests which followed no noticeable change in this noise was noticed. The attachment was left in this position since it only offset our null travel position by 0.083 inches.

TABLE III. NO LOAD OPERATION TIMES

	#1	#2
Wrist Watch - Together	15 sec up	16 sec up
Wrist Watch - Together	17 sec up	17 sec up
Stop Watch - Together	First up	1.65 sec later
Stop Watch - Together	First down	1.0 sec later
Stop Watch - Up Alone	15.6 sec	17.0 sec
Stop Watch - Down Alone	12.9 sec	13.8 sec

- The following tests on #2 were conducted sequentially in an exploratory manner to see why the plow stalled at 1-1/2 inches off the floor:

  - No load exercises up to latching were run a few times. Tank battery power at 26-1/2 volts was used.

  - As a quick load availability application, four men at approximately 800 pounds stood on the plow. The plow rose over 15 inches and was stopped for safety reasons. Tank power at 26-1/2 volts was used. The initial motor case temperature was 84°F and 20 volts/44 amps were recorded.

  - The unloaded plow was raised 3 inches off the floor to bypass the 1-1/2 inch stall point and the 35 percent load condition was applied. Prime power was switched here, and for the rest of the tests in this sequence, to a power module set at 28-1/2 volts. The plow raised several inches from the 3 inch start. The temperature rose from 89°F to 95°F and 50 amps were recorded. No voltage was recorded.

  - The preceding test was rerun but with a loaded start from 2 inches above the floor. Again the plow rose several inches. The temperature rose from 99°F to 102°F, and 24 volts/45-50 amps were recorded.

  - The 35 percent load was then attempted to be raised directly off the floor. The plow rose to the 1-1/2 inch point and again the overload lamp came on. The temperature rose from 107°F to 109°F, and 65 amps were recorded. No voltage was recorded.

The preceding three tests seemed to confirm that there was enough motion resistance in the #2 pushbeam connection to the mounting bracket to cause the overload to function at its particular setting.

  - Finally, the 42-1/2 percent load weight was placed on #2 plow after it was raised 6 inches off the floor. The plow raised to 10-1/2 inches and the overload lamp came on. The temperature rise was 120°F to 122°F, and 23 volts/65 amps was recorded.

Observations throughout the functional tests of the overload circuits were noted to be both time and current sensitive as designed. Short trip out times at high current as well as longer trip out times as lower currents were noted.

The marginal performance of the present motor was determined to be aggravated by certain accumulative electrical system losses not originally anticipated but were ingrained into the system because of design constraints. The E-MALS system wiring was selected on the basis of voltage drops when

conducting a nominal 50 ampere or less current projected for the lifting loads and from the limiting hole size in that vehicle's headlamp adaptor. The actual current required was found to be much higher in these tests. These higher currents, their duration, and repeated cycling quickly overheated the motor windings, raising their resistance, which caused the motor to operate even slower, which in turn lowered the actuator efficiency. Concurrently, the system line voltage drop, relay contact voltage drop, and the connector pin voltage drops became larger than the designed voltage drops. These accumulative losses throughout the E-MALS system acted in concert to quickly establish a degenerate performance process. Future design considerations should take into account this degenerate feedback effect on performance when component ratings are specified.

It was finally decided that to ensure that the testing at APG not be hampered by the overload cutout that it be raised as close to the 50 percent field test value as possible. Also, that this setting be accomplished without the moldboard and skid shoes in order to allow the plow motion to pass through the horizontal attitude of the pushbeam. The ensuing tests were limited in loading due to availability of weight masses.

- Plow #1: 3,035 pounds at 62.13 inch moment arm were loaded to produce 46.5 percent of maximum load. The plow rose 3 inches off the floor in about 3-1/2 seconds when the overload lamp came on; 25 volts/65 amps were recorded.

- Plow #2: 3,035 pounds at 56.8 inches were loaded to produce a 43.0 percent load. The plow rose 10 inches off the floor (no stall occurred at the 1-1/2 inch point) before the overload lamp came on. No volts or amps were measured.

This last test concluded the qualification test program. The TWMP system was stripped from the vehicle, packaged, and with the spare parts, shipped to Aberdeen Proving Grounds.

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6.0 CONCLUSIONS

The concept feasibility demonstration E-MALS delivered to APG had a 50 percent lesser load capacity than that specified in the design duty cycle. This degradation is a result of the optimistic assumptions for the dynamic efficiency of the gearbox coupled with the reduced capability of the motor in an overheated condition. The in-house functional and qualification tests have concluded however that the E-MALS concept is a viable solution for the replacement of the electrohydraulic lift system. Sufficient design knowledge has been gained to enable a development design to proceed with confidence.

7.0 RECOMMENDATIONS

In order to make the E-MALS design a more reliable approach for plow lifting the general recommendation is to increase the performance ratings of the gearbox and motor, to provide a means to dissipate the heat generated by the motor, to provide more effective waterproofing seals, and to revise and improve the control system.

The mechanical portion of the E-MALS concept is fairly straightforward and the recommended changes which have resulted from the fabrication and test of the concept demonstration model can be integrated with relative ease. The concept demonstration model however has one serious deficiency in that it is a one unit assembly which requires complete removal from the push beam for repair or replacement of any of its components. It is recommended that the next generation of the E-MALS concept adapt a modular design wherein the ball screw actuator, gearbox, motor, or controls can be removed and serviced independent of each other.

The design analysis discussions of the E-MALS control system, section 4.1.4, was based upon conceptual needs of the system as originally anticipated. The feasibility of a TWMP system operating with an E-MALS unit and incorporating automatic controls and protective circuits were the principal objectives. As a result of the tests conducted in-house, and subsequently at APG, it appears that certain controls (speed and position) are superfluous. In particular, the option of using full UP power with partial DN power as opposed to reducing power near the limit stops was found to be the better compromise of these options, which are preset by switches on the plug-in boards. It is recommended that this option be deleted in any future models in favor of a fixed full UP-partial DN power operation. The IC circuits thus remaining may be gainfully used to correct other shortcomings which will be described later. The test and operations of the TWMP also provided the opportunity to further evaluate the DN cycle operation which was very difficult to predict analytically. During the heavy load lifting tests it was realized that the actuator unit starting friction was much higher than its dynamic friction when it was observed that the motor labored at start-up with reduced power based upon DN commands. Presuming that the

motor DN speed would quickly become destructive at full power during plow lowering, the initial design provided for reduced power in this mode and this reduced power was automatically applied, due to the circuit characteristics, in the reverse direction (UP). This start-up problem was corrected on the demonstration model control box by incorporating an override switch into the circuit which overrides all normal control circuits and executes the order of the UP-DN switch immediately without any other priority (overload, etc.). This switch could also be used in the event of failure of any of the automatic circuits, but the operator must always assume the responsibility for UP-DN stops and excessive speed, current, and motor temperature. It is intended that only a momentary action of this switch will start the motor DN cycle and will offset any actuator start loads. Because of the high risk which can occur with frequent use of this switch (the remainder of the control system is fully automatic), it is recommended that future models incorporate a DN automatic start circuit.

Various load and plowing tests conducted have readily verified the need for the present automatic current based overload control circuit. These operations have also shown that with a stalled actuator the rise in motor input current, converted to heat, quickly becomes destructive to the motor components and most likely to other circuit components, including needless fuse blowing. The current overload protection system did not protect the demonstration model motor from heat destruction when subjected to continuous rapid re-start since the current overload circuit resets to zero each time the operator cycles the control switch.

To protect the motor against thermal overload it is recommended that future models incorporate a premotor-stall circuit based on rpm off the potentiometer and a thermal overload circuit that would also actuate the present current protector circuit. The thermal overload circuit would sense motor winding temperature and automatically indicate an overload and shut the system down to a cooloff point. In a tactical emergency situation the operator could still operate the E-MALS system with the override switch.

The rate at which the motor cools down or the time required for the thermal overload lamp to go off is a function of the motor thermal characteristics, the immediate heat paths to the surrounding components, and environmental factors. It is recommended that in future models design considerations should be given to the heat dissipation problem. Specifically, the motor should have a more direct contact with the actuator housing which can serve as a heat sink and the motor waterproofing cover should have cooling fins. Using a higher rated motor would help a great deal since it would operate with reserve capacity and its heat generation would be less.

A summary of the recommendations to improve the basic E-MALS concept are listed below in order of importance:

MANDATORY (FUNCTIONAL)

- Select a larger motor and gearbox (casting)
- Revise the actuator unit for waterproofing
- Add heat dissipation paths
- Increase the system wiring size
- Incorporate an automatic down start-up circuit
- Incorporate a pre-stall motor circuit based on rpm off the potentiometer
- Incorporate a temperature overload circuit
- Change the single fuse concept to separate fuses
- Change the power control option to full UP and partial DN (This is actually a simplification change)
- Modify the electronic boards for printed circuit (PC) construction with slide mounts and potentiometer adjustments

COST REDUCTION (NON-FUNCTIONAL)

- Divide the large control box into a control panel and an auxiliary box to include ballast resistors
- Delete use of the headlamp connector and relocate the mounting bracket terminal board to the headlamp
- Change the electrical diodes to I. C. type on the P. C. board

SAFETY - MAINTENANCE - CONVENIENCE

- Break up the actuator unit into a modular design
- Add locks to the override switch and interlock ON-OFF switch with the master POWER switch
- Include a travel lock limit switch and indicator for automatic operation
- Provide a calibrated dial potentiometer for presetting the plowing depth by the operator
- Relocate the actuator grease fittings for easier access
- Specify ball screw with end stop to prevent over extending
- Specify proper housing interfaces with gaskets for sealing

APPENDIX A

PURCHASE DESCRIPTION - MOTOR

The motors for the concept demonstration model were obtained from the Electric Motor Division of Robbins and Myers, Inc. and were specified by the following purchase specification.

PRELIMINARY TWMP MOTOR SPECIFICATIONS

1.0 General

1.1 Motor, electrical, direct current, series 28 volt, 1-1/2 hp, 10,000 rpm, 50 amps

1.2 Materials

Unless otherwise specified herein, materials shall conform to Drawing No. TD137760 (R&M 750200301)

1.3 Construction

Construction and assembly shall conform to Drawing No. TD137760 (R&M 750200301)

2.0 Mechanical

2.1 Motor shall be open frame, ball bearing with 4-12" leads for reversing and fan cooled.

2.2 Motor shall be flange mounted, with 7/16 keyed shaft and maximum outline dimensions of Drawing No. TD137760 (R&M 750200301)

3.0 Electrical

3.1 Motor shall be for intermittent service at both 28 volts and 14 volts. Motor load cycles are shown in Figure A-3.

3.2 Motor speed torque curve shall be as in Figures A1 and A2 at 28 volts.

3.2.1 Motor torques shall be $\pm$ 20% in the forward direction - CCW facing motor shaft. Reverse torques may be 50 percent less.

3.3 Motor shall produce 1.25 hp minimum at 10,000 rpm and 28 volts.

3.4 Motor shall mate with the following load cycle at 28 volts; 8 seconds at 60 oz. ft.; followed by 21 seconds at 12 oz. ft.

3.5 Motor shall be capable of repeating the cycle of 3.4 every five minutes.

3.6 Motor shall provide 1,000 rpm minimum at 60 oz. ft. at 28V.

4.0 Environment

4.1 Motor must meet the shock and vibration requirements of MIL-810-B as follows:

4.1.1. Shock method 516, Figure 516-1, Procedure 1, 40 G's, 18 MS

4.1.2. Vibration method 514, Procedure VIII, Curve W, Figure 5, Parts 1, 2, and 3.

4.2 Temperature

Motor must meet the performance cycle of 3.5 while enclosed in a metal container with a 250 cubic inch volute of captive air. The initial ambient temperature of the motor mounting flange and the air about the container shall be +125°F to -25°F.

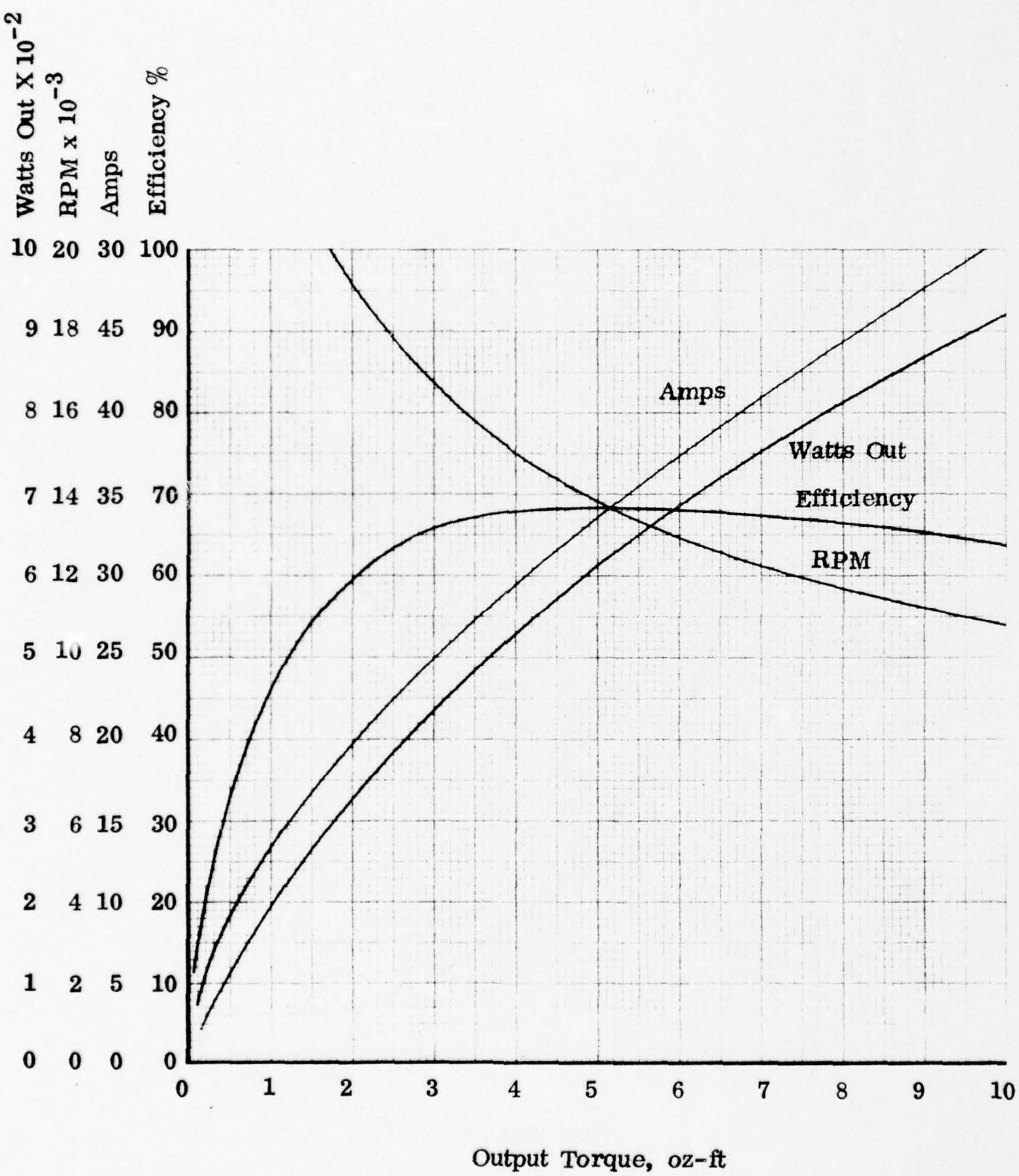

Figure A-1. Nominal Performance

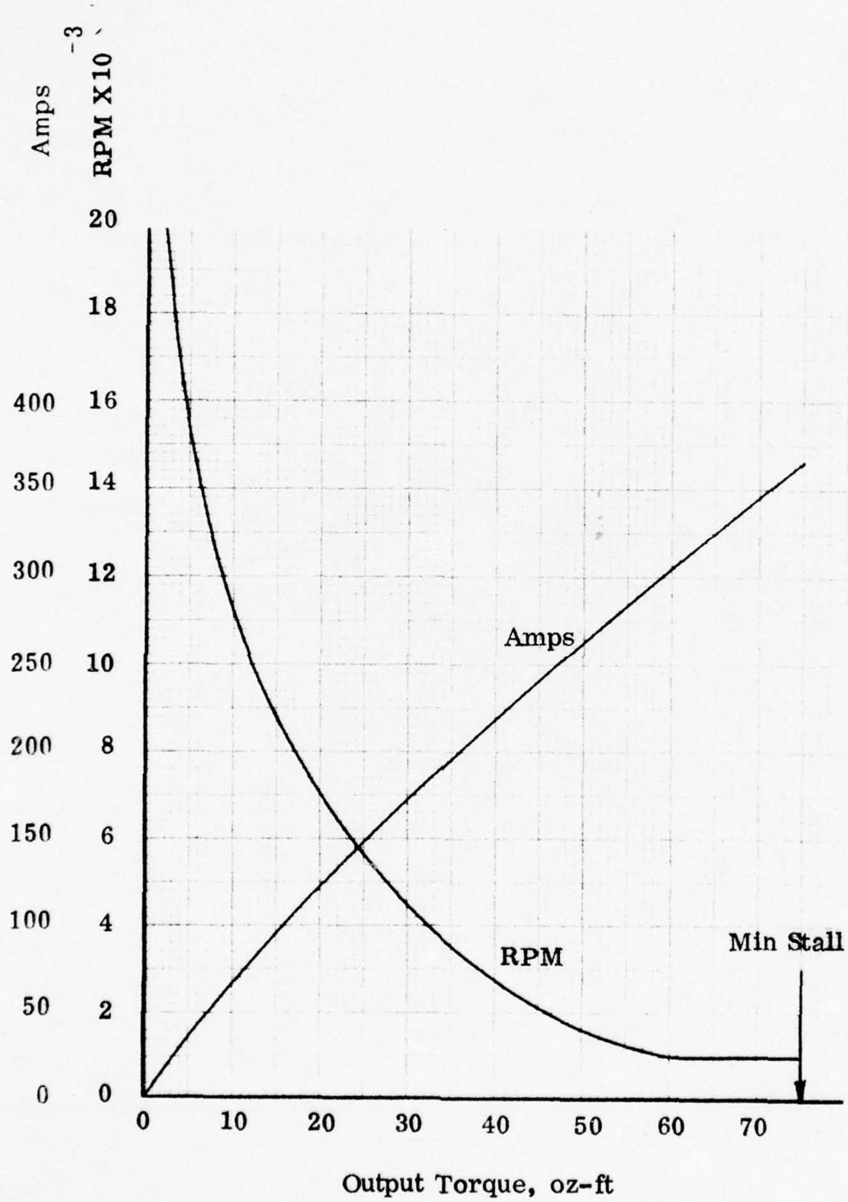

Figure A-2. Nominal Performance

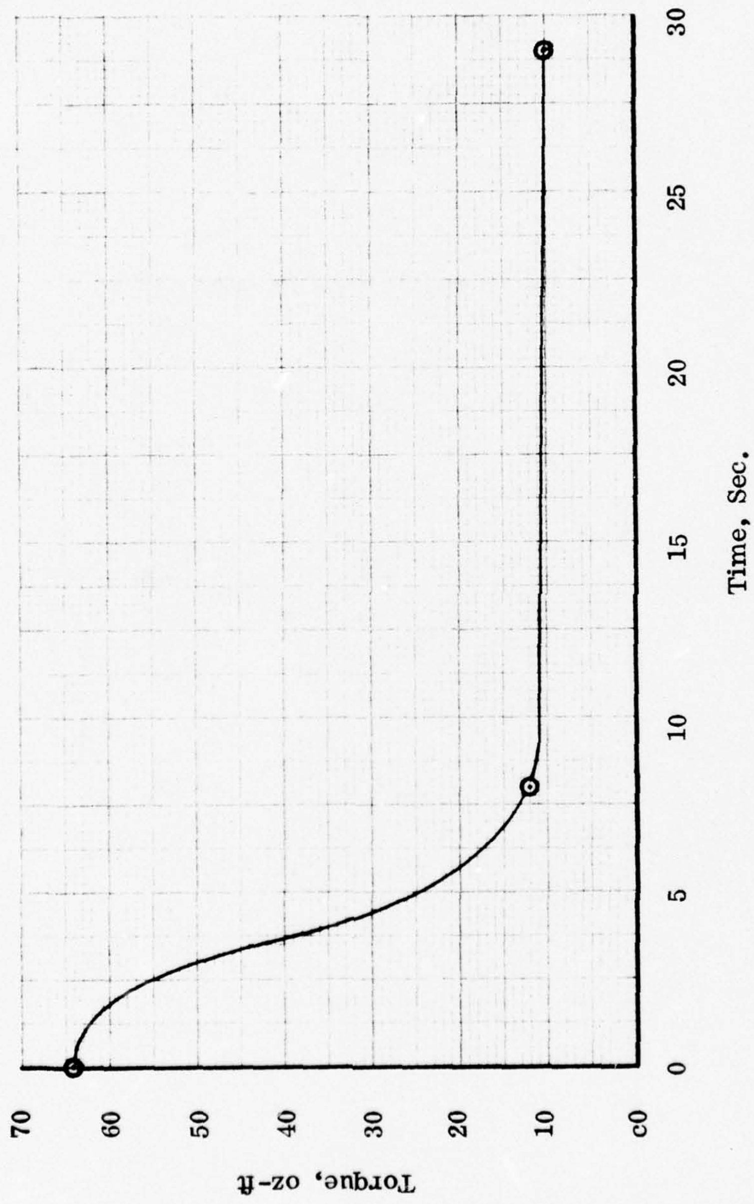

Figure A-3. Typical Load Profile

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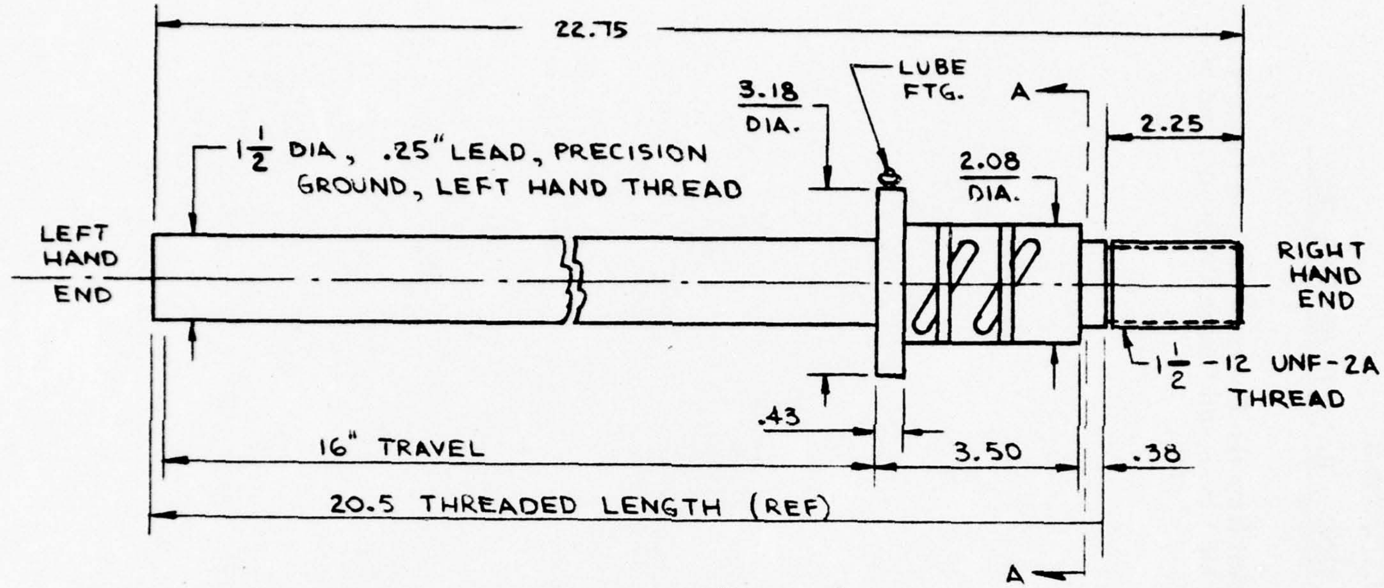
APPENDIX B

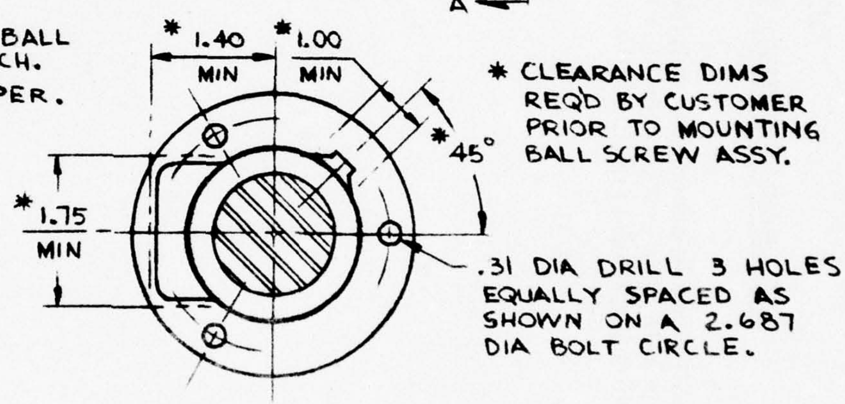
PURCHASE SPECIFICATION - BALL BEARING SCREW

The ball bearing screw was obtained from the Beaver Precision Products, Inc., a subsidiary of Warner Electric Brake and Clutch Company, and was specified by the data on the following sketch.



- SINGLE NUT MODIFIED TO 2 BALL CIRCUITS OF 3 1/2 TURNS EACH.
- INTEGRAL HARD STEEL SCRAPER.
- .011 MAX BACKLASH.
- .006 IN/FT ACCURACY.
- INTERMITTENT DYNAMIC LOAD CAPABILITY APPROX 35,840 lbs.
- 17-4PH SS CONSTRUCTION.

BEAVER PRECISION PRODUCTS
*B-17302 REF W24-75 7-29-75



SECTION A-A

APPENDIX C

TWMP PLOWING LOADS ANALYSIS

1.0 INTRODUCTION

A review of the various failure reports of the TWMP during D. T. Phase II testing has instigated a program to produce a NASTRAN structural model for the plow hardware. A necessary initial input to the formulation of this model is a definition of the magnitude, direction and points of application of the external plowing loads for the possible modes of vehicle plowing operations.

The primary objective of the loads analysis was to derive in an expedient manner the maximum steady state loads (no oscillatory loads, etc.) which theoretically could be imposed on the plow. Modes of operation were to include straight-ahead plowing on level ground, large turning maneuvers, side slope plowing, trench crossings and oblique plowing into embankments. Additionally, an impact type of loading (as striking an immovable rock) was included which would occur during any of the steady state conditions. As the analysis progressed it became evident that those resultant load vectors which became critical could be generated with reasonable similarity by only a select few of the operational modes. With this knowledge, the considerations of extensive NASTRAN computer time with all the possible load applications, the loads analysis spectrum was reduced to three operational modes; case 1, steady state level ground plowing with full throttle tank stalled by one plow in an embankment; case 2, steady state level ground plowing and one plow impacts an immovable object; and case 3, steady state level ground plowing with an ineffective skid shoe whose load is transferred to the lift chain.

Although preliminary loads analyses were made of each of the possible operational modes and delivered to the NASTRAN modelers, this report will only describe the analysis approach and results of the selected three cases.

2.0 APPROACH

The determination of plowing loads were made by four avenues of approach or combinations thereof:

1. A review was made of TWMP test data collected by APG and MERDC in 1972 which utilized an M728 vehicle with an early hardware configuration of the plow. The plow's raising hydraulic cylinder, its turnbuckle crossbeam and its skid shoe pin were instrumented with strain gages to measure loads. Also the vehicle's final drive torques, sprocket RPM and vehicle speeds were also measured. Discrete data points were recorded at .0025 second increments for each of 49 test runs. This test program produced a wealth of oscillatory type loads values which was useful for fatigue analysis but proved to be a futile approach (considering out limited task time) in determining maximum steady state external plowing loads at the selected load application points. This approach had some value however in that a quantitative feel was obtained as to the relative magnitude of forces distributed within the instrumented plow structural members.

2. The observation of test movies made of the plow in operation through various types of terrain dramatically depicted the violent oscillatory and somewhat random motions of the plow and dirt movement. These observations revealed that in reality any "steady state" plowing forces at fixed input application and direction points were superimposed by random oscillatory load vectors. When the results of this loads analysis are given later in section 3.0 they should be accepted with the true picture of "steady state" in mind.

3. A stress analysis of the failed components would reveal, by tracing back through the load paths, the input load vectors which caused the failure. This approach in determining plowing loads necessitates the use of the NASTRAN results which are predicated upon the plow input load vectors. Clearly this approach is quantitatively useful only in checking the results of the NASTRAN model, using analytical input loads, against the actual failure modes. This check, in fact, is programmed by the recipients of this load analysis task. The failure modes, however, did provide a qualitative insight as to the type of load vector input which could conceivably cause such a failure. This approach aided in the selection of initial load points and in culling out those operational modes which would overtax the computer time allocation.

4. A theoretical evaluation of the input plow loads can be made using the mobility performance characteristics of the M60A1 vehicle which are well documented. This approach formed the basis of the loads program and is described in detail in section 3.0. The various assumptions, selections of load points and vector directions and operational modes used in the theoretical approach were conditioned by the observations discussed in items 1, 2, and 3 above.

3.0 ANALYSIS AND RESULTS

The drawbar (DB) push, or pull, of the M60A1 vehicle is defined as the resultant force available for work, either to propel the vehicle at some speed on level ground, allow it to climb slopes, or to pull or push a load. The net DB force is obtained from the vehicles tractive effort produced at the drive sprockets after motion resisting forces due to tracks, wind, road (terrain), and track slip losses are subtracted out. The net DB force is usually plotted against the vehicle speed for the mobility conditions specified. This plot indicates the maximum producible speed of the vehicle with the net DB force acting against it. As the DB force is reduced the vehicle speeds up and as the DB force increases the vehicle will slow down to a point where it cannot move anymore. This is called the stall condition and represents the maximum DB force produced by the vehicle's powertrain.

It can readily be seen that if the vehicle's DB force was applied to push a plow then the total plowing resistance force on the plow is the DB force. With this basic premise, the completion of the load analysis required a logical assumption as to the points of plow-dirt load interface and the distribution of the total DB force between these points.

Figure C-1 is a plot of the nominal M60A1 drawbar performance on level cross-country terrain with no track slip and operating in the first transmission gear. These data were generated by an established computer program which utilized TACOM generated terrain resistance loads. The 100 percent throttle curves and the no slip conditions were selected for the loads analysis in order to represent the maximum DB force conditions possible. Pertinent values used include 73,238 pounds DB at the stall condition and 29,613 pounds DB at 5 mph.

While plowing forward, the plow is apparently loaded on all of its forward face. For simplicity these myriad of load points were assumed to be consolidated into six resultant points: At the center face of each tooth, at the center of the moldboard, and at the skid shoe pin. These then are defined as the load points for the "steady state" DB resistant loads. Now considering that a plow can strike an immovable object, the foremost plow points are the tip of the skid shoe, the tip of any of the teeth, the outboard edge of the moldboard and conceivably the center of the moldboard between the teeth. These latter points are defined as the "impact" load points, to be discussed below. These forward moving vehicle plow load vectors are assumed to act unidirectionally along the moving axis with any lateral or vertical components generated only by the geometry of the plow.

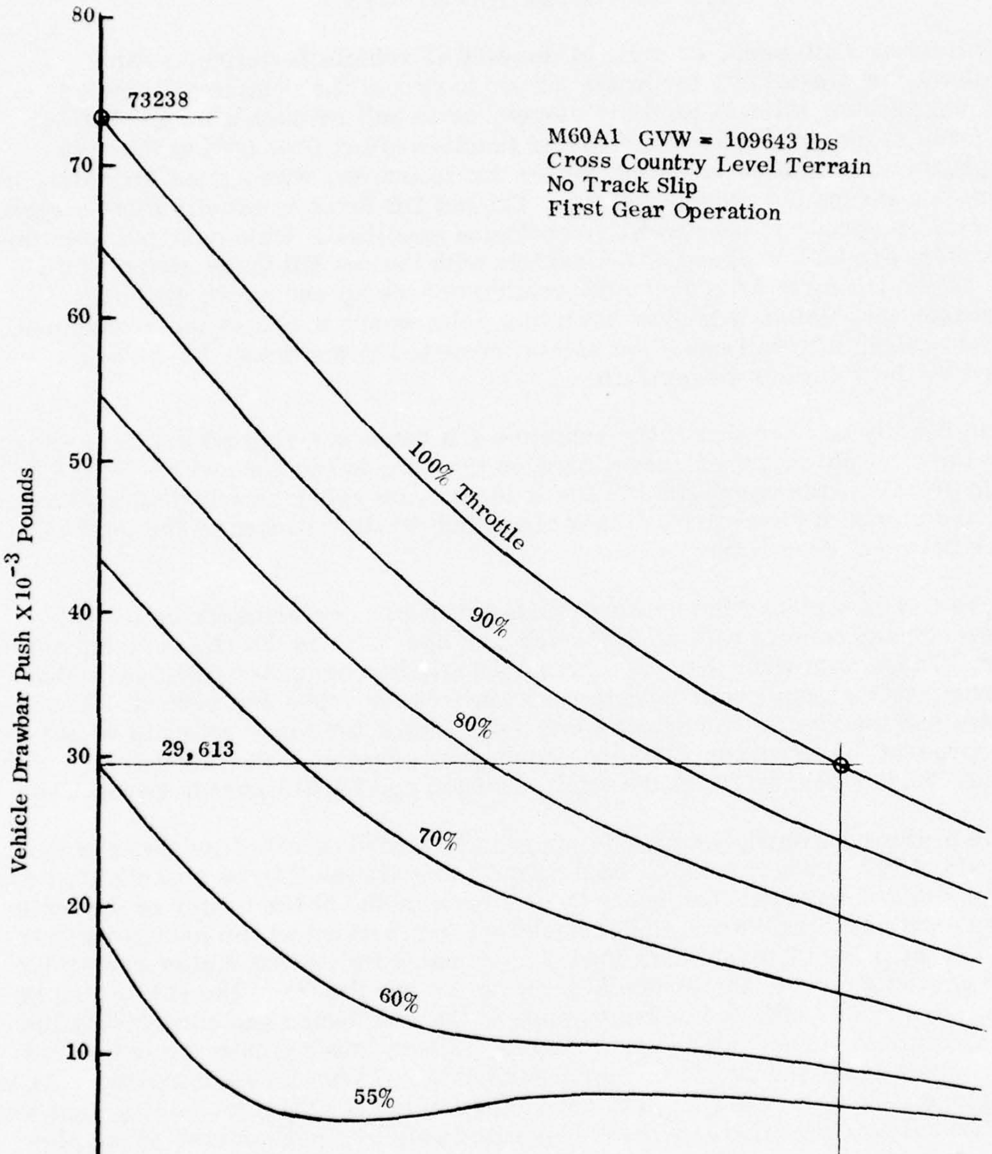


Figure C-1. Computer Predicted M60A1 Drawbar Push vs. MPH

The load vectors will change if the vehicle operates in other than a straight line mode and in addition may alter the point of load application. These various other possible operational modes of plowing such as turning, in and out of a trench, side slope and down slope along with possible impacts were given a preliminary computation to indicate the effect of the changes in the axial DB load vectors. The load vectors do change and in some instances increase such as when the vehicle's weight component adds to the DB force in down slope operations or when the vehicle pitches into a trench or the vectors shift on the tooth when turning. A subsequent review of these data by the NASTRAN modelers resulted in their selection of the three modes delineated in Section 1 which, in all cases, has the vehicle operating in a straight-line forward motion on level ground.

The distribution of the horizontal axial DB steady state force resistance on the plow between the six selected application points can be determined from a static load diagram of the plow, figure C-2. In this diagram, the load points are numbered 1 through 6. Point 1 is the center of the moldboard, 2, 3, 4, and 5 are the centers of the tooth faces and 6 is the skin shoe pin. The horizontal vector is designate x, the vertical vector is z and the lateral vector is y. It is assumed that each tooth axial and vertical vector magnitudes are the same:

$$F_{2x} = F_{3x} \approx F_{4x} \approx F_{5x} = F_{2z} = F_{3z} \approx F_{4z} = F_{5z}$$

and that $F_{6x} = \mu F_{6z}$ where μ is the soil to skid shoe coefficient of friction. Taking moments about the hinge pin point 0, plus the summation of horizontal and vertical forces produces:

$$F_{6z} = \frac{.0972\ F_{1x} + 5.0187\ F_{2x}}{(1-.2271\mu)} \tag{1}$$

$$F_{6z} = \frac{DB-4F_{2x} - F_{1x}}{\mu} \tag{2}$$

$$F_{6z} = 4\ F_{2x} + R_o \tag{3}$$

$$\text{Where } R_o = DB\ (\tan 9.5°) = .1673\ DB \tag{4}$$

Combining equations (1) through (4) and assuming $\mu = .5$ we obtain:

$F_{1x} =$ .51659 (DB) on the moldboard

$F_{2x} =$ .06662 (DB) Horiz. and Vert. load on each tooth

$F_{6z} =$ .43383 (DB) Vert. skid shoe pin load

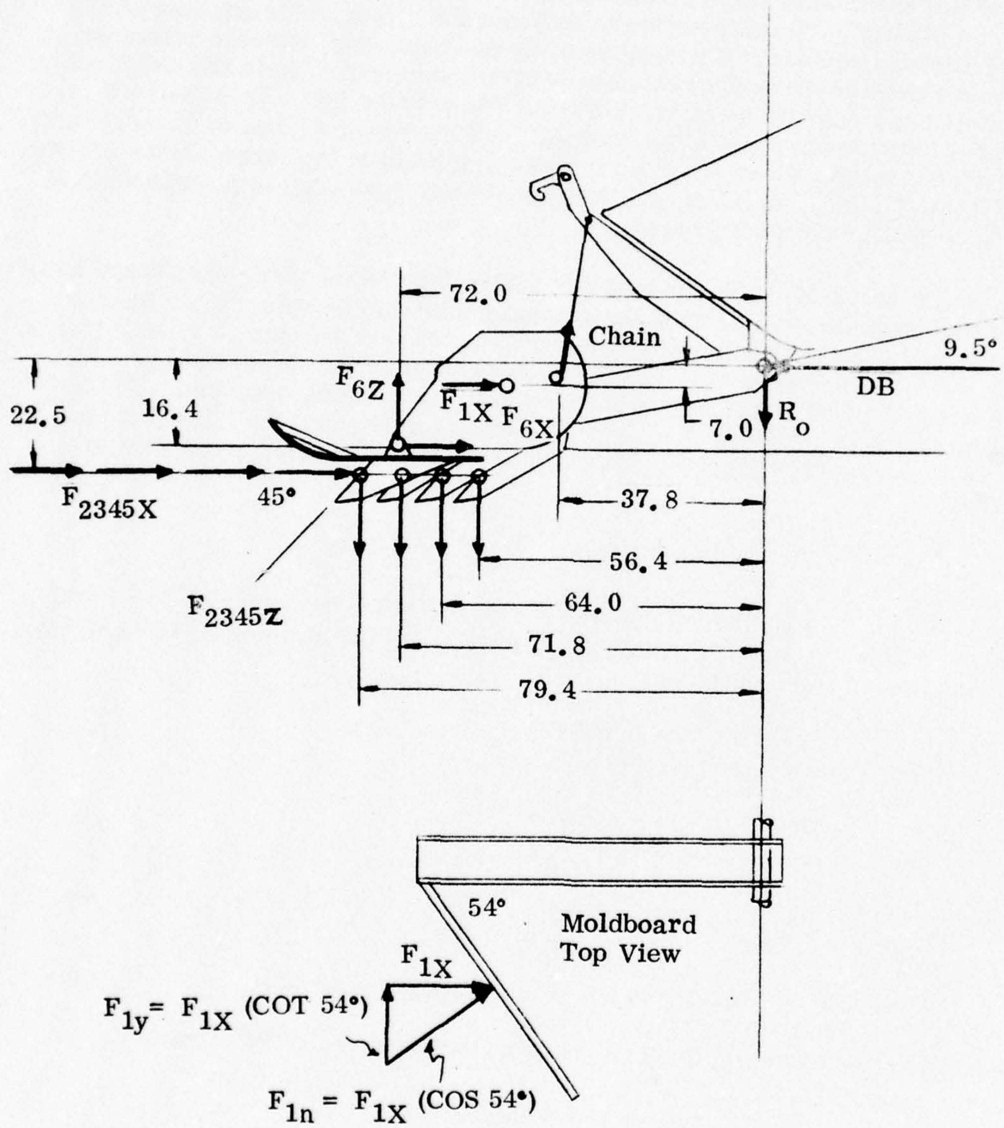


Figure C-2. Static Load Diagram

$F_{6x} = \ .21692$ (DB) Horiz. skid shoe pin load

from which it is seen that the axial DB load is distributed as approximately 51.7 percent on the moldboard, 26.6 percent on all the teeth and 21.7 percent at the skid shoe pin. Thus for any selected value of DB force input by the vehicle the load values on the selected application points can be determined.

The plow moldboard is horizontally tilted approximately 54° from the vehicle's longitudinal axis. From the diagram on Figure C-2 it is determined that the normal force on the moldboard is:

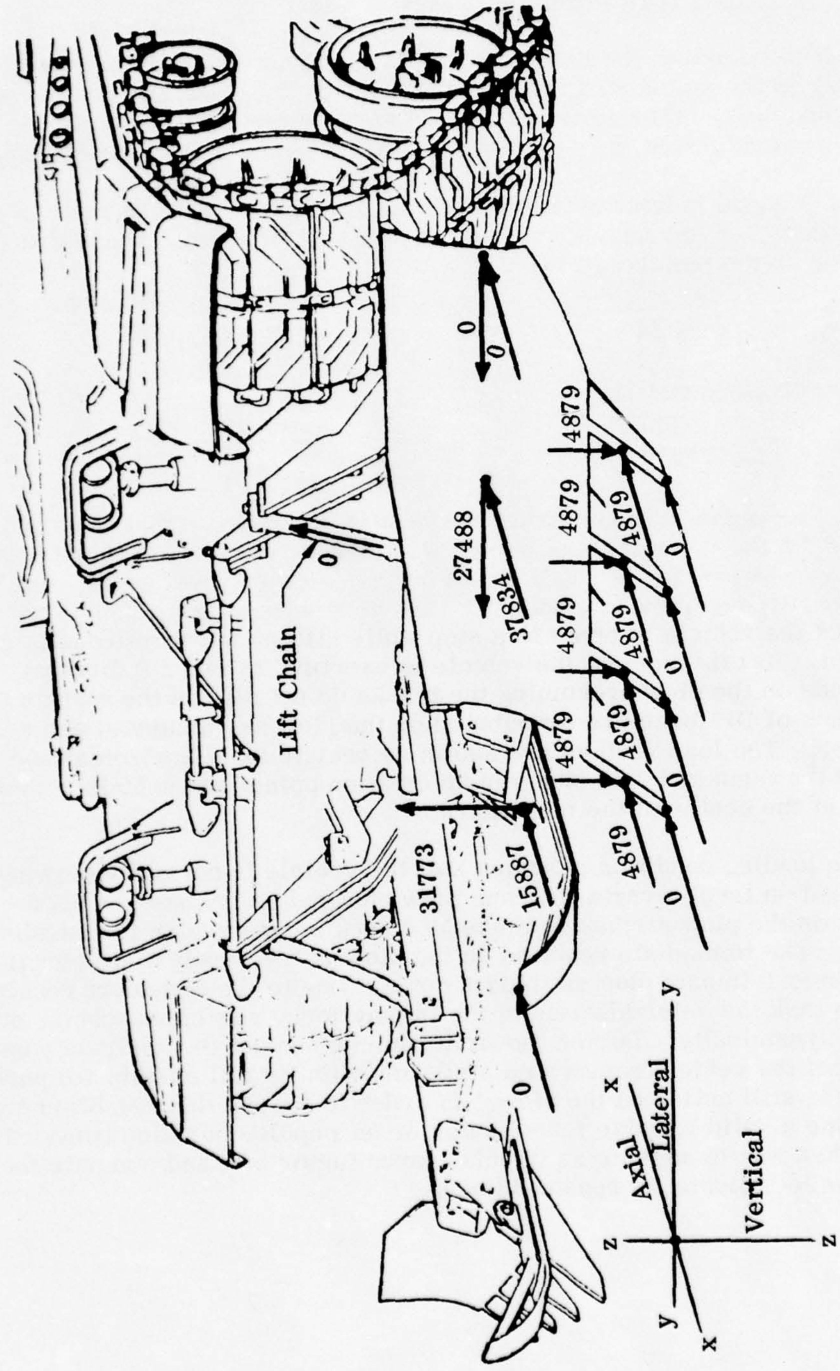
$$F_n = F_{1x} (\cos 54°)$$

and the lateral component is:

$$F_{1y} = F_{1x} (\cot 54°)$$

We are now in a position to construct the first load condition case (Case 1). In order to determine the maximum steady state plow loads on a plow we assume that the vehicle is plowing straight ahead in first gear on level ground and for some reason only one plow is engaged. This plow encounters a bank of earth which causes the vehicle to come to a stop while still at full throttle first gear power output. At this moment the vehicle is exerting its full DB force of 73,238 pounds on the plow presuming the tracks do not slip on the terrain. The 73,238 pounds of DB force are distributed on the plow as calculated and shown in figure C-3. The load vectors are shown as vehicle axial horizontal and vertical for the teeth and skid shoe pin application points and vehicle horizintal and lateral at the center of the moldboard.

The impulse loading condition assumes that the vehicle is normally plowing straight ahead on level terrain with one plow engaged as in case 1 when a select point on the plow strikes an embedded rock or post which is considered immovable. The immediate reaction on the plow is the steady state plowing load it had before impact plus an impact force. Due to the expedient requirement of this task the determination of the impact force was approached statically rather than dynamically. During the finite time of impact the analysis approach presumes that the vehicle comes to a stall stop with its full no-slip DB push of 73,238 pounds still acting on the plow. In order to assess this condition one must presume a valid impulse force factor or an impulse duration time. The approach taken was to assume an impulse force factor of 2 and compute the impulse time to validate its reasonableness.



Mode: Level ground plowing, one plow engaged, vertical plow load resisted by skid shoe. Plow encounters resistance causing vehicle to stall and exert its maximum drawbar push.

Figure C-3. Case 1 Plow Loads

Ignoring the transient force values during the impulse time and the time to stall the vehicle the simplified static approach is to presume that the vehicle was initially plowing at 5 mph with 29,613 DB force on the plow. This leaves a DB force potential of 73,238-29,613 = 43,625 pounds to stall the vehicle. Presuming that this 43,625 pounds is supplied by the rock then the impulse force is 2 x 43,625 pounds = 87,250 pounds. This value is conservative, of course, since the full 43,625 pounds is not developed until full vehicle stall which is caused in turn by the 43,625 pounds of rock resistance. In summary, the impulse force loads are defined as the dual application of the steady state axial horizontal load computed as a distributed function of the 5 mph DB push as in case 1, plus the 87,250 pounds of impact force which is presumed to also react in the horizontal axial direction at any one selected impulse point.

As a check on the impulse duration time we note that impulse is equal to mass x velocity and also to force x time:

$$\text{IMP} = \frac{W}{g} V = Ft, \text{ lb-sec}$$

from which

$$t = \frac{WV}{gF}, \text{ sec}$$

At W = 109,643 pounds of vehicle weight

V = 1.4667 (5 mph) = 7.3335 ft/sec during time t

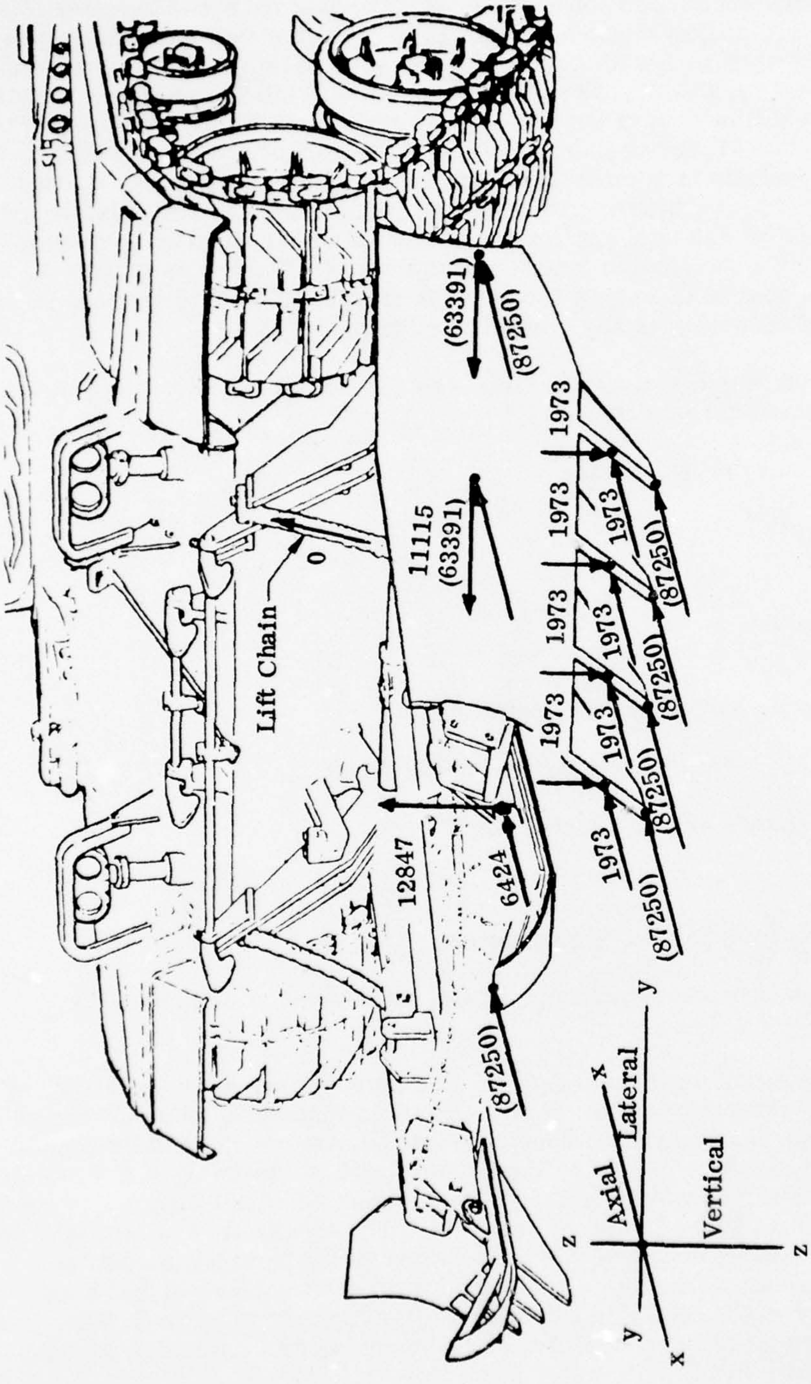
F = 87,250 pounds average during time t

we obtain:

$$t = \frac{(109643)\ (7.3335)}{(32.2)\ (87250)} = .286 \text{ seconds}$$

which appears to be a reasonable impulse reaction time considering the approximations used.

Figure C-4 depicts the results of the impulse plow load analysis, Case 2. Care must be exercised in reading this figure since the impact force alone is presumed to act only at one select impact point at one time. The select impact points are the tip of the skid shoe, the tip of any of the four teeth, the outboard edge of the moldboard, or the center of the moldboard. The diagram gives the impulse force at all these points in brackets. The steady state 5 mph plowing loads is shown without brackets and these loads do act simultaneously as discussed in the Case 1 analysis. Note that only at the center of the moldboard can one directly vectorily add the impulse force to the steady state load. The lateral loads on the plow moldboard are computed as a function of the axial load as per Figure C-1.

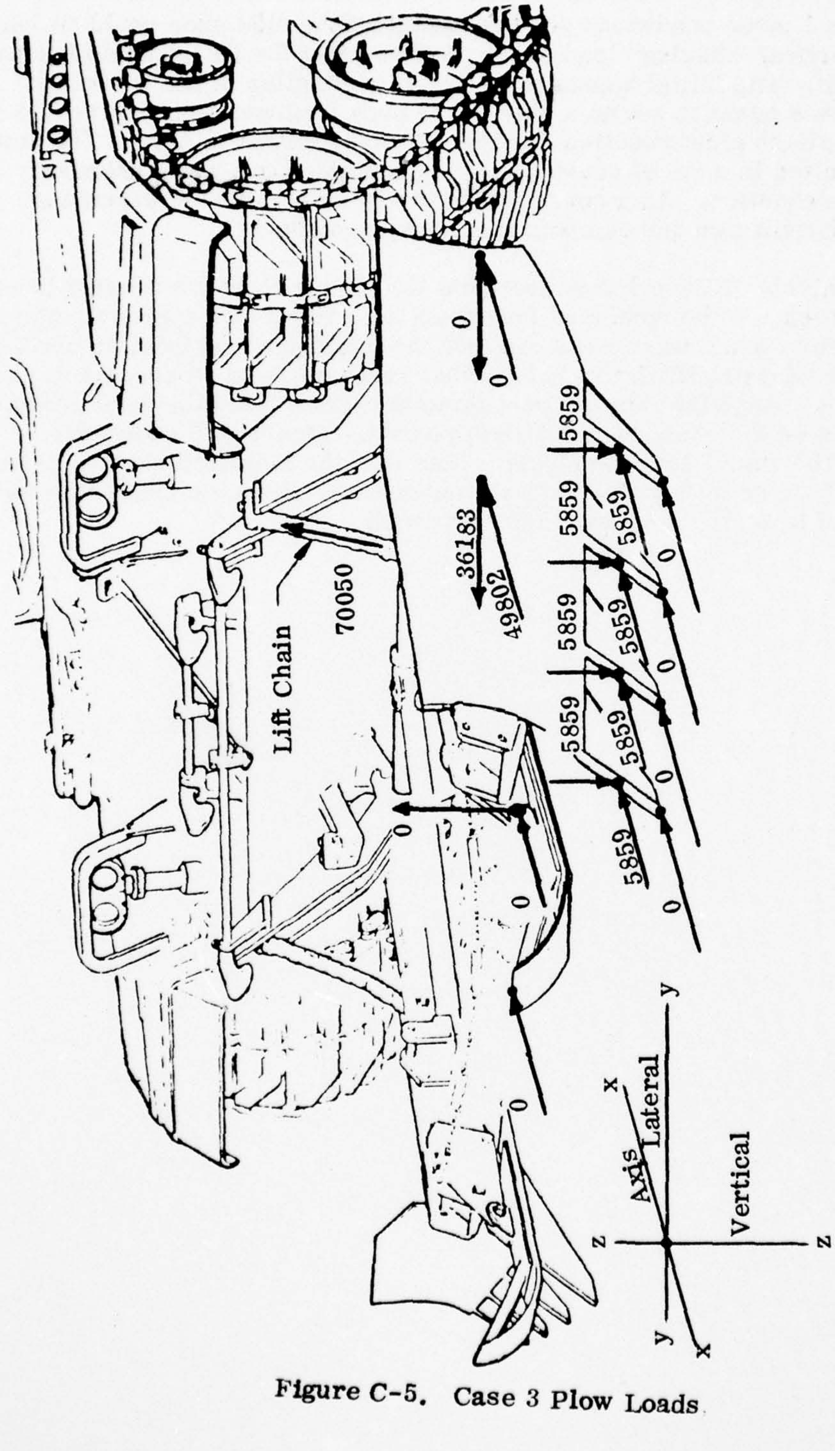


Mode: Level ground plowing at 5 MPH, one plow engaged, vertical load resisted by skid shoe. One point on plow hits a rock with a zy impulse force of 87250 lbs. Other points on plow loaded per 5 MPH drawbar push (impact load).

Figure C-4. Case 2 Plow Loads

The third case of plow loading by direction of the NASTRAN modeler was to take the case 1 mode conditions and presume that the skid shoe could no longer resist the vertical "digging" load of the plow and that the lifting chain has taken over this load. The initial approach to the determination of the DB force distribution was again to set up a static free body diagram as in figure C-2 but with a chain attach point reaction and the skid shoe forces at zero. The static diagram resulted in a set of equations which contained one more unknown variable than equation. As a consequence it was determined to assume a reasonable distribution and compute the loads accordingly.

From the analysis of Case 1 it is seen that the plow moldboard roughly takes twice the DB load as the combined four teeth do. Using this approach, the load distribution for Case 3 was to assume that the moldboard resisted 68 percent of the horizontal axial DB force and the four teeth resisted 32 percent (8 percent on each tooth). Applying this assumption to the static force diagram produced a tension load on the chain of .9565 (DB) pounds. Figure C-5 shows the summary of the case 3 loads analysis. Note that the indicated chain load of 70,050 pounds is not a vehicle vertical load but acts along the chain axis which was computed to be 78.5° up from the horizontal.



Mode: Level ground plowing, one plow engaged, vertical plow load resisted by lift chain. Plow encounters resistance causing vehicle to stall and exert its maximum drawbar push.

Figure C-5. Case 3 Plow Loads

4.0 CONCLUSIONS

The load values given for the three cases are considered to be quite conservative considering the assumptions of no track slip, full vehicle stall power, immovable rocks, "static" impulse computations, and concentrated load application points. They do, however, fulfill the objectives of this task to provide a preliminary input to the NASTRAN structural model. Caution must be exercised by those wishing to utilize these load values for other purposes in that their derivation and assumptions must be understood.

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APPENDIX D

TWMP STRESS AND NASTRAN ANALYSIS

NASTRAN Model - M60A1 Mine Plow

A structural analysis of the M60A1 Mine Plow was made using the NASTRAN finite element program. The static analysis capability of NASTRAN was used to determine internal loads, stresses, and deflections throughout the plow structure.

The model is a three dimensional representation of the plow primarily using plate elements which transmit in-plane forces, in-plane shear, and bending forces. The major portions of the moldboard, pushbeam, and mounting bracket are modeled using plate elements. The bolts attaching the moldboard, skid shoe, and diagonal cross brace to the pushbeam are beam or bar elements. The aft portion of the pushbeam, the diagonal cross brace, the lateral cross tube, and the mounting pins are also fashioned out of bar elements. The structural elements are located and connected together by an array of grid points. A schematic representation of the basic model is shown in figures D1 - D3.

Analysis is accomplished by applying a set of load vectors representing a previously-determined load condition to appropriate grid points on the plow teeth and moldboard. See appendix C for these load conditions. Support for the structure is obtained by constraining the proper grid points on the vehicle towing lug where the mounting pin attaches and on the lower portion of the mounting bracket which bears against the vehicle hull. Reactions to the applied load act through these support points to balance the structure. Support locations are shown in figures D4 and D5.

For a given loading on the structure, NASTRAN generates internal loads distribution, stresses in plate and bar elements, deflection of grid points, and reaction forces at constrained grid points. These results are then reviewed to locate or verify problem areas. The program does not use material yield and ultimate strength values and hence does not recognize possible failure areas as they occur. This is accomplished during the review of the results. Due to complexity and time limitations, no progressive failure analysis was undertaken.

Evaluation of Baseline and Field Mod Plows

As a basis for comparison, the plow prior to DT II field modifications was modeled and evaluated for several of the load conditions as defined in appendix C. A second model was made incorporating the DT II fixes which included mounting bracket reinforcement, pushbeam front-end modification, removal of small mounting pin, and addition of a center support to the lateral cross tube. Because of the large number of load conditions and the limited time the comparison was limited to case 1: straight and level plowing with all drawbar load on one plow. NASTRAN results for the two plow configurations are shown in figures D6-D14. Front end of pushbeams: figures D6, D7, & D8 show this problem area and principal stresses for before and after beam reinforcement. For 1020 steel, the ultimate bending allowable is approximately 70 ksi. Adding the reinforcement lowers the maximum bending stress from 93 usi to 62 ksi on the side plate and from 160 ksi to 59 ksi on the upper plate. Figures D9 and D10 show the maximum principal stresses for the mounting bracket plates. It is seen that reinforcing the bracket results in substantial reduction of the stress levels. A more important consideration for the mounting bracket is the lateral deflection causing the bracket to take an S shape when viewed from the front of vehicle. Figures D11 and D12 show the NASTRAN results for lateral deflection at the bracket grid points before and after addition of reinforcement. A reduction in deflection by a factor of ten is evident verifying both the necessity and adequacy of the structural reinforcement. The S shaped deflection for before and after addition of reinforcement is shown in figure D13. This lack of stiffness of the mounting bracket side plates was a contributing factor to mounting pin failure which occurred when the lateral motion of the plates caused the mounting pin restraint to fail and allow the pin to work itself part way out of the attachment. For load case 1, the bending stress in the center of the lateral cross tube dropped from 77 ksi to 24 ksi with the addition of clevis-type center support. These results are shown in figure D14.

Quick Release Mounting Bracket

Effort was expended in an attempt to define a quick release technique for the TWMP. This effort included both design and supporting structural analyses.

The initial design effort consisted of the generation of a number of conceptual designs. These concepts covered several feasible approaches to each of the more critical areas of concern: Push beam to hull interface and pin configuration, center beam to hull interface and type of disconnect for both structural and electrical systems. Proper load path control was mandatory.

Of the several concepts generated, two were selected by the Government for further design development. Full size layouts were generated of both

concepts and as the designs matured one appeared to offer more advantages than the other. This concept was sufficiently completed to establish feasibility of the approach. All indications were that the design was feasible and practical.

It employed two explosive bolts in a redundant arrangement such that firing these bolts resulted in the immediate release of the entire plow. All load paths were established and structural adequacy established by the use of computer modeling.

Such a study using NASTRAN was performed to determine the effects on internal loads of pinning vs. fixing the ends of the diagonal cross brass and the aft end of the pushbeam. It was found that the most satisfactory internal load distribution resulted for vertically-oriented pins at both ends of the diagonal cross brace and at the pushbeam connection to the lateral cross tube.

To aid in design of the plow with quick-release mounting bracket, a NASTRAN model of the preliminary design was made. Results of this model dictated structural changes to be incorporated into the design.

When the design was complete, a verification model picking up all the changes was made and run.

Using the NASTRAN generated stresses and internal loads, a comprehensive analytical review of the design was made.

The design and analysis of the Quick Release version of the TWMP was not pursued further by direction of the Government.

Stress Analysis in Support of Plow Modification

For M60A1 Vehicle

To prevent lift chain failure, a new upper attachment to the mounting bracket providing a swivel was designed. The structure of the new attachment was analyzed to be as strong as the lift chain in tension. It was necessary to make a 1.0 inch diameter hole through the main cross tube in the vicinity of the pushbeam to accommodate the wiring harness for the electric actuator. The bending moment in this section of the tube was obtained from the NASTRAN results for case 1. The stress was then calculated for the section before adding the 1.0 inch hole and found to be 59 ksi and after adding the 1.0 inch hole to be 90 ksi. A hardness check of the actual tube determined that its bending strength is above 90 ksi and hence it is of sufficient strength with the 1 inch hole. A slot had to be made in the U-shaped pushbeam connector to accommodate the harness. The primary plowing loads act on the pushbeam in compression and do not load the connector making this modification structurally acceptable.

Normal plowing (case 1) load condition was used to redesign the lateral cross tube restraint or center clevis. Designing to this load level resulted in a thinner, lighter structure than the one previously designed.

For M728 Vehicle

A swivel-type upper lift chain attachment was designed for the M728 plow mounting bracket in a similar fashion to the one designed for the M60A1 plow.

Use of a center clevis to restrain the cross tube on the M728 plow is not possible because of the great distance from the tube to the vehicle hull. An alternate approach is to add reinforcement to increase the bending strength of the tube. The size of the reinforcement was determined using the following approach: 1) determine bending moment that will cause ultimate failure in existing tube, 2) design reinforcement such that yielding first occurs at the above moment, 3) determine best orientation of reinforcement by reviewing NASTRAN loads. Provisions for assembling the plow diagonal struts to the cross tube dictated a design somewhat stronger than that calculated above.

Since plow side load component is transferred to the vehicle at the mounting bracket-vehicle lug interface it is recommended that the large gaps between the lug and bracket sides be filled with shims. Based on the NASTRAN results for the M60A1 plow mounting bracket reinforcement, it is recommended that similar reinforcement be added to the M728 mounting bracket to the extent possible.

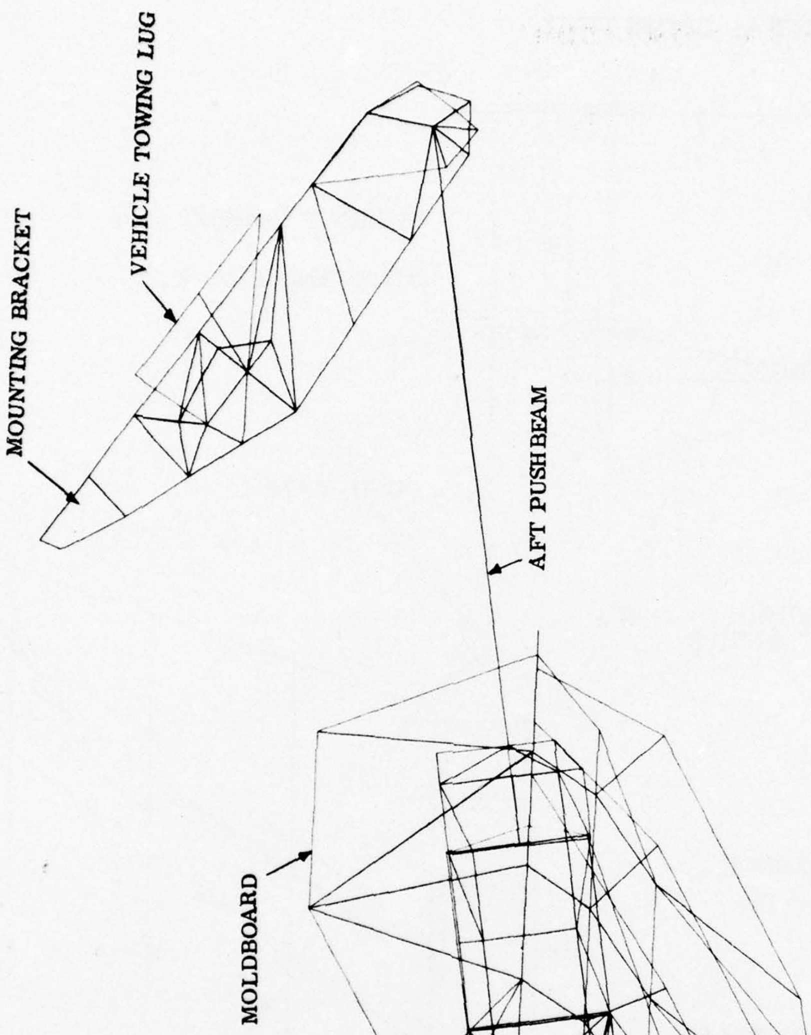

Figure D-1. Side View of Nastran Model

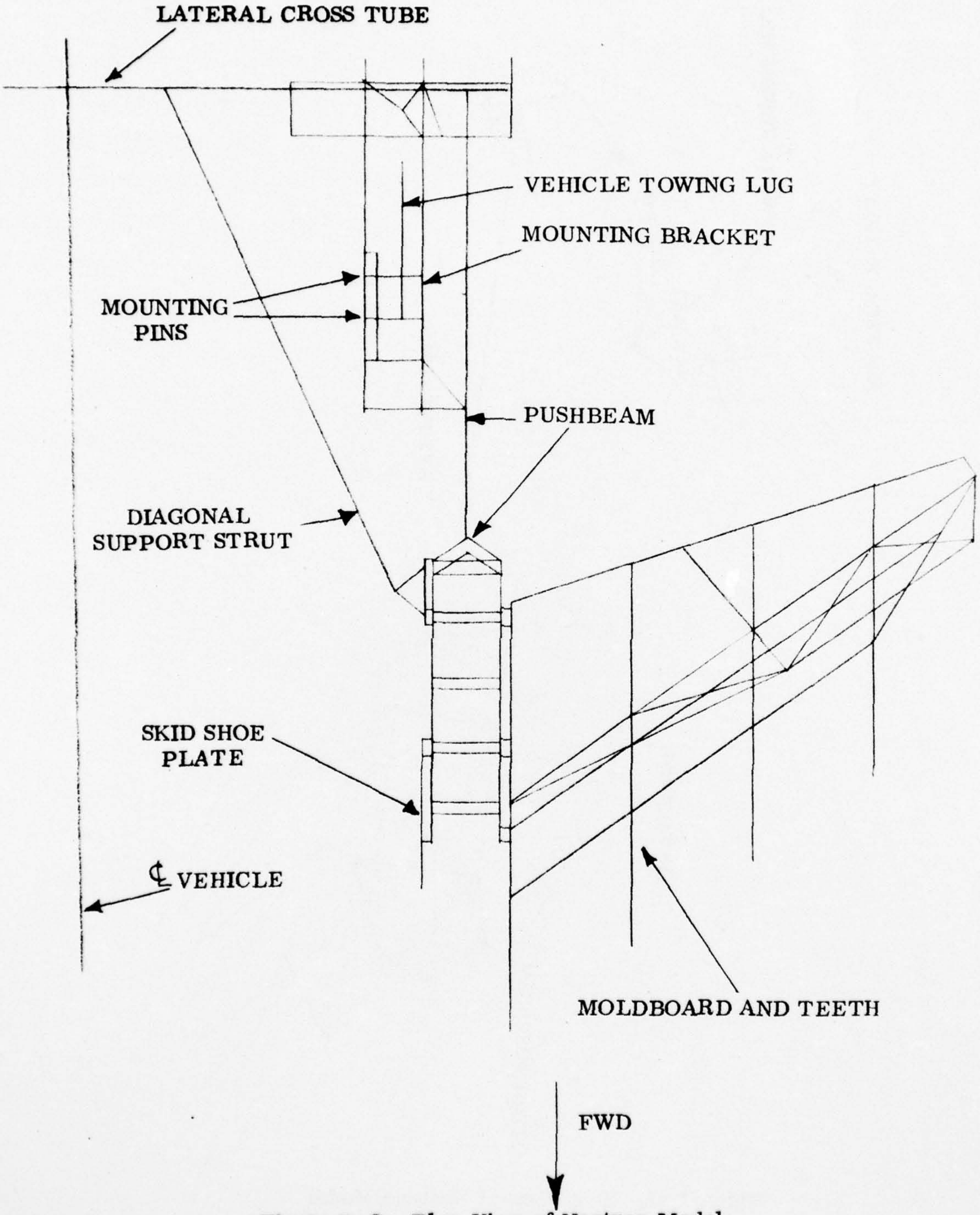

Figure D-2. Plan View of Nastran Model

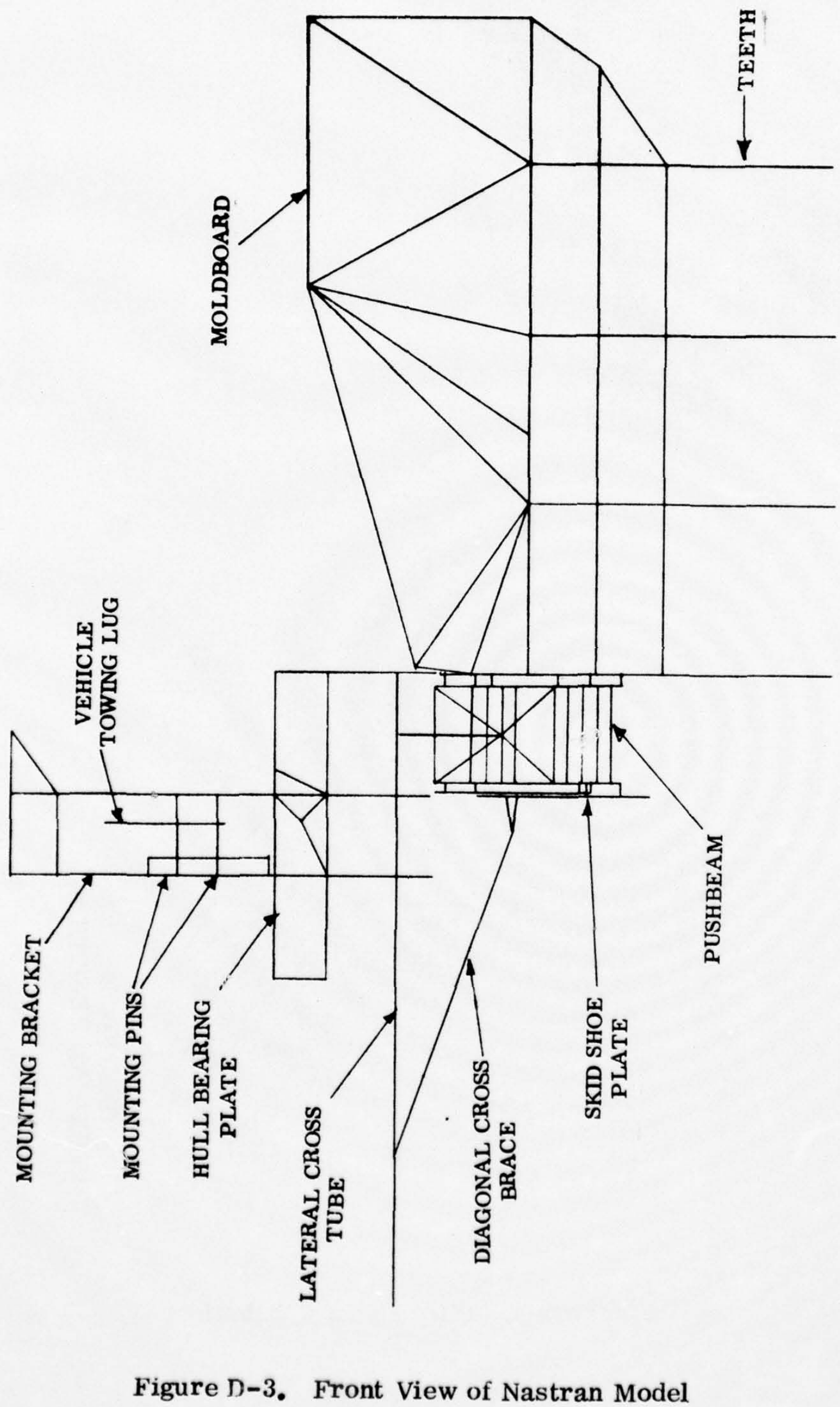

Figure D-3. Front View of Nastran Model

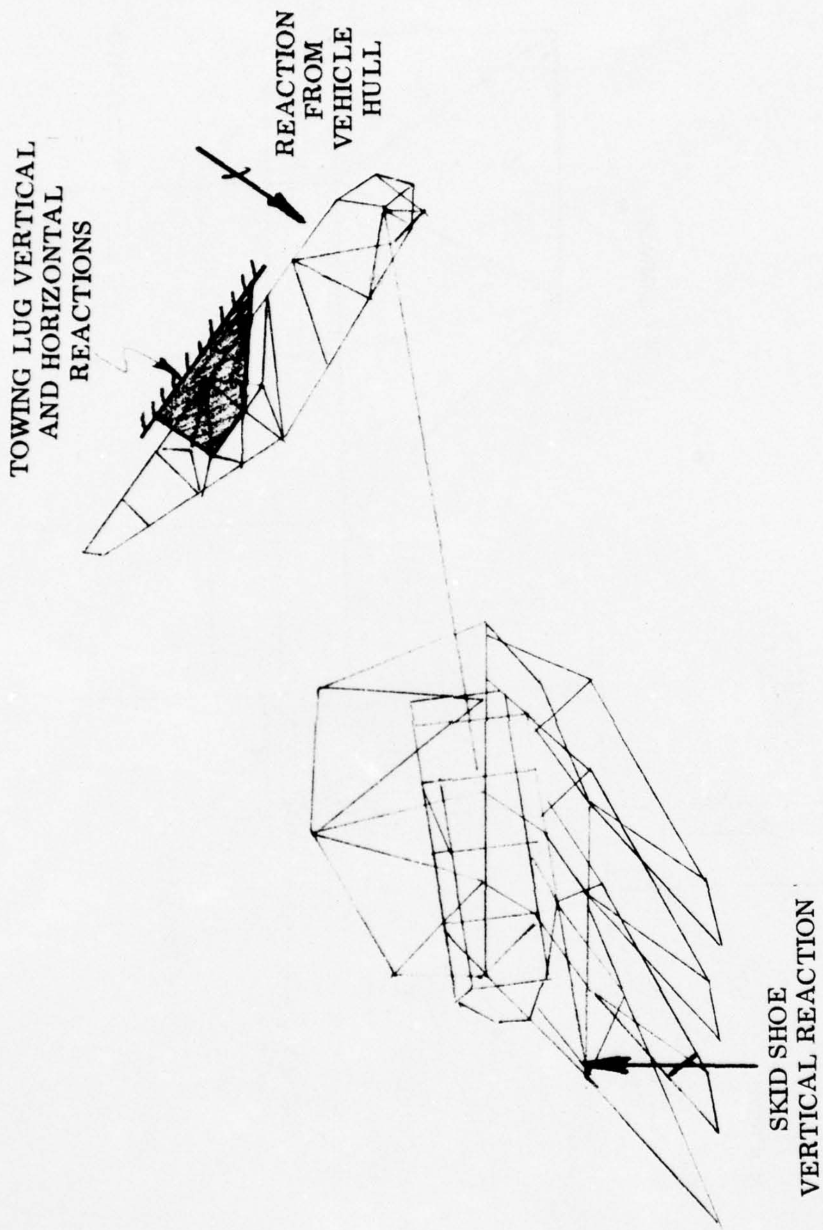

Figure D-4. Plow Model Constraints

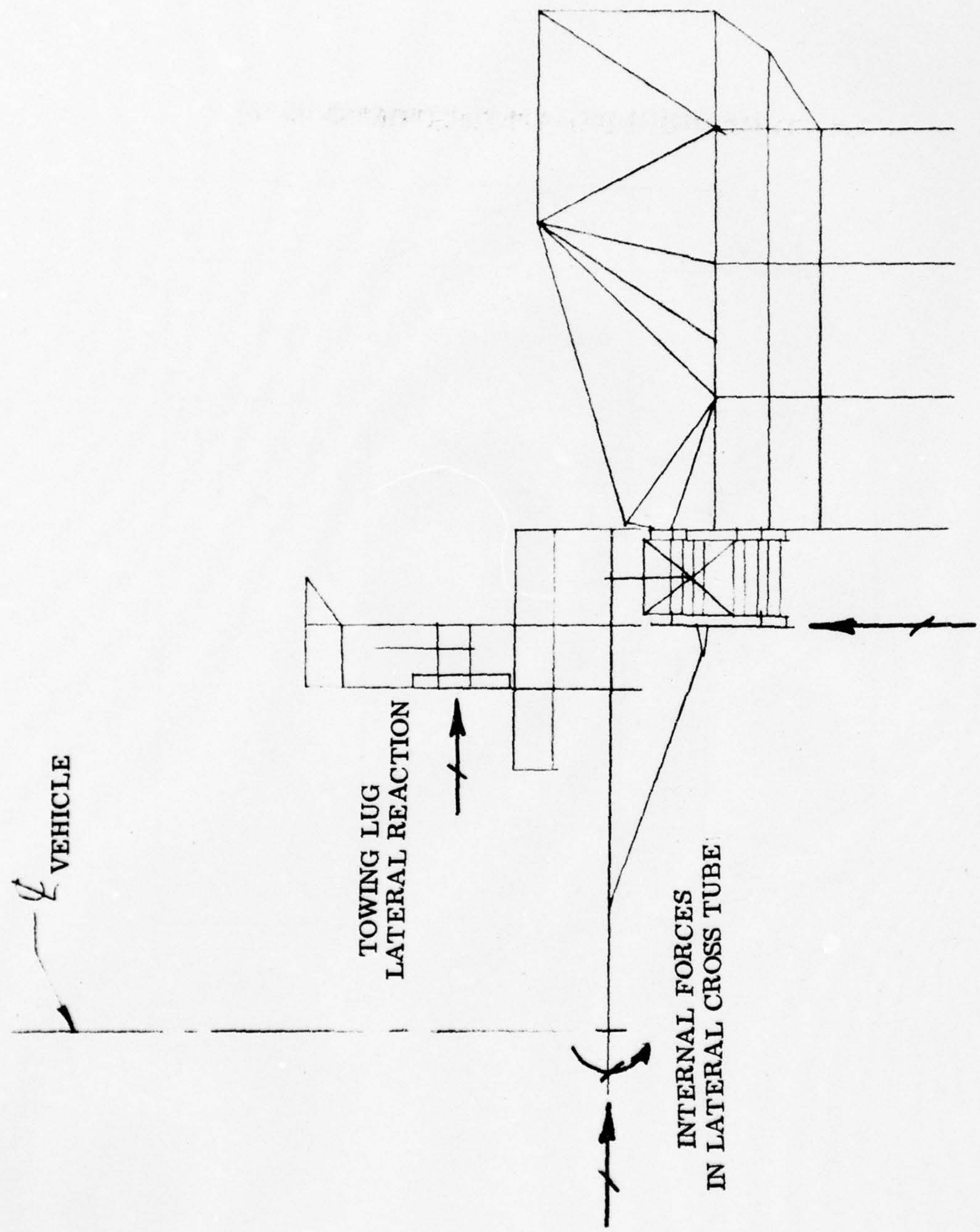

Figure D-5. Plow Model Constraints

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ELECTRO-MECHANICAL LIFT SYSTEM VERSION OF THE DT II TRACK WIDTH--ETC(U)
JUN 76 J D KUBINA, M GRENUS, C J CHRISTMANN DAAK02-72-C-0451

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MAXIMUM PRINCIPAL STRESSES IN KSI

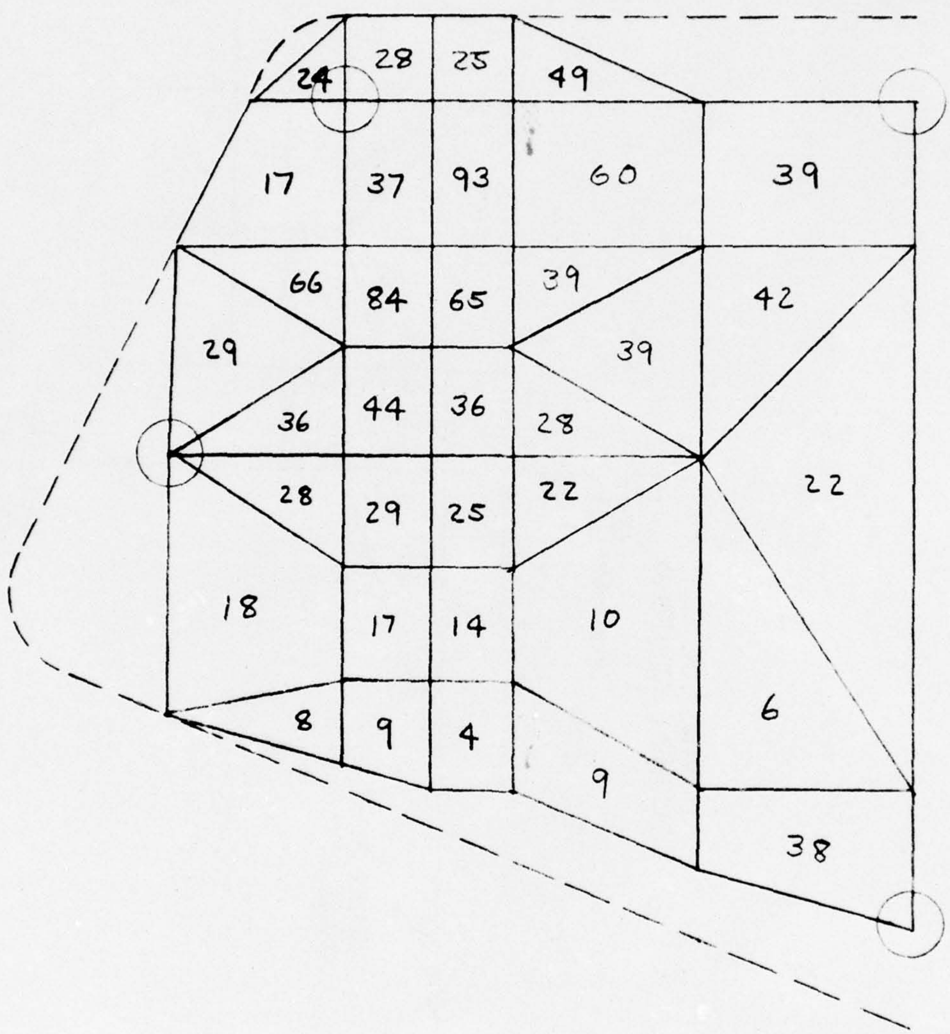


Figure D-6. Pushbeam Stresses Baseline Plow Model

MAXIMUM PRINCIPAL STRESSES IN KSI

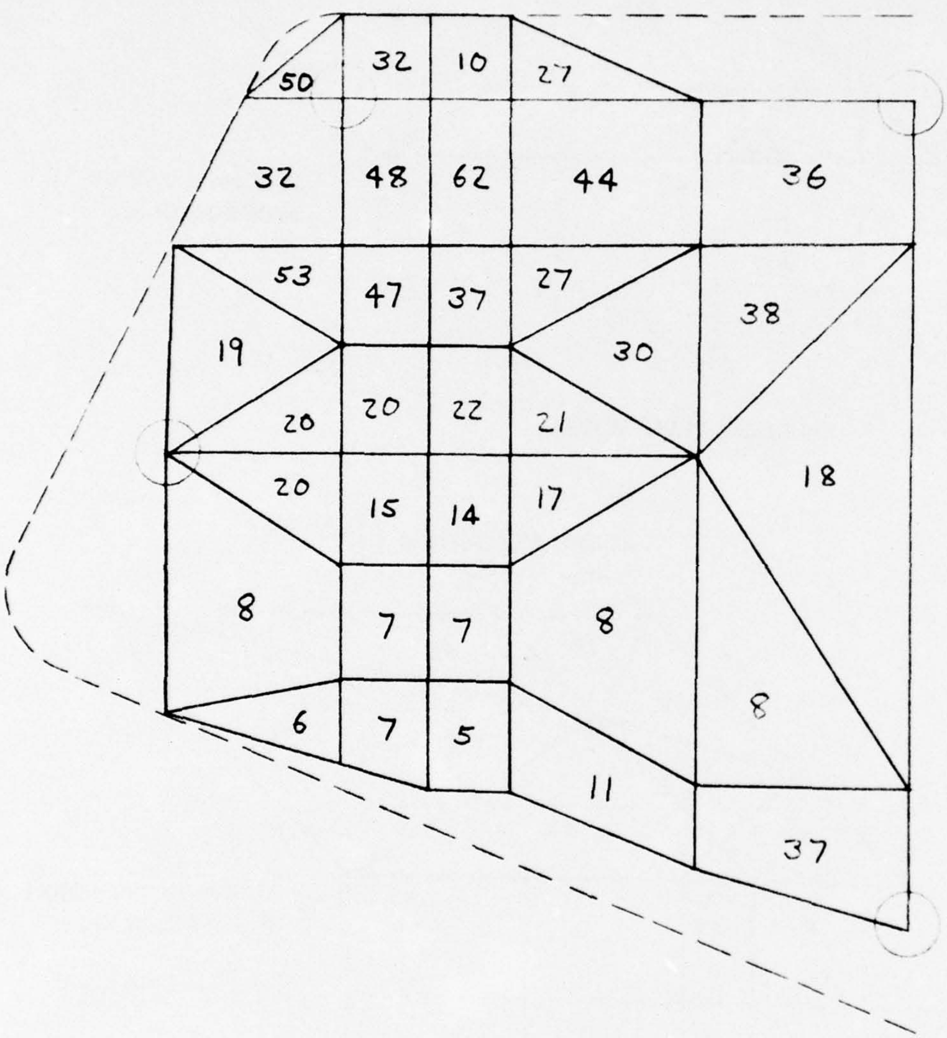
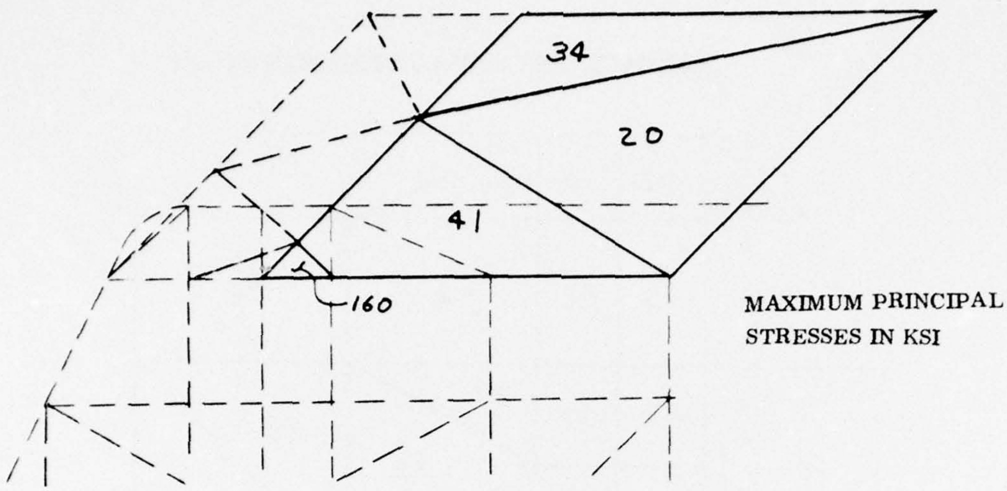
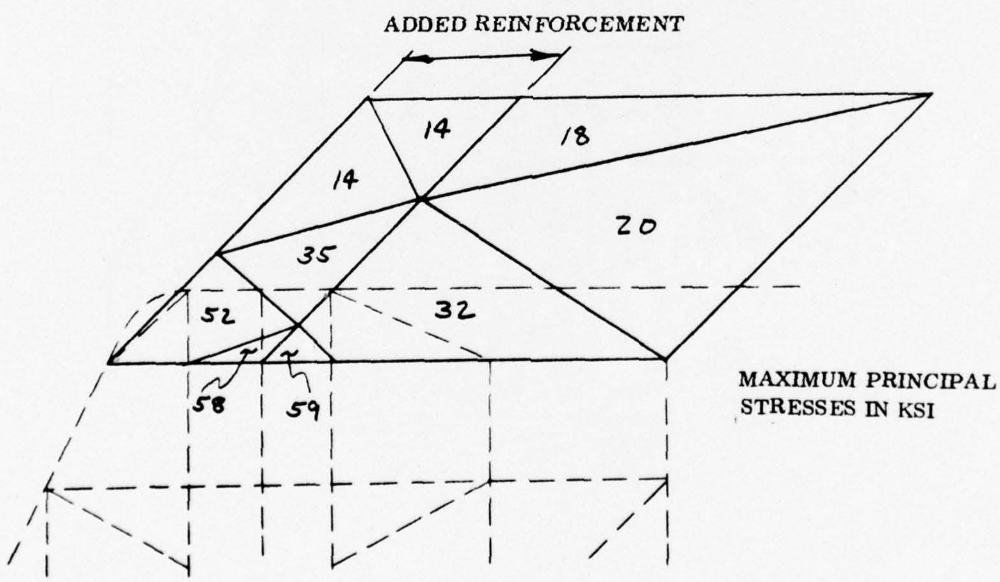


Figure D-7. Pushbeam Stresses After Addition of Reinforcement



BASELINE PLOW MODEL



AFTER ADDITION OF REINFORCEMENT

Figure D-8. Comparison of Pushbeam Stresses

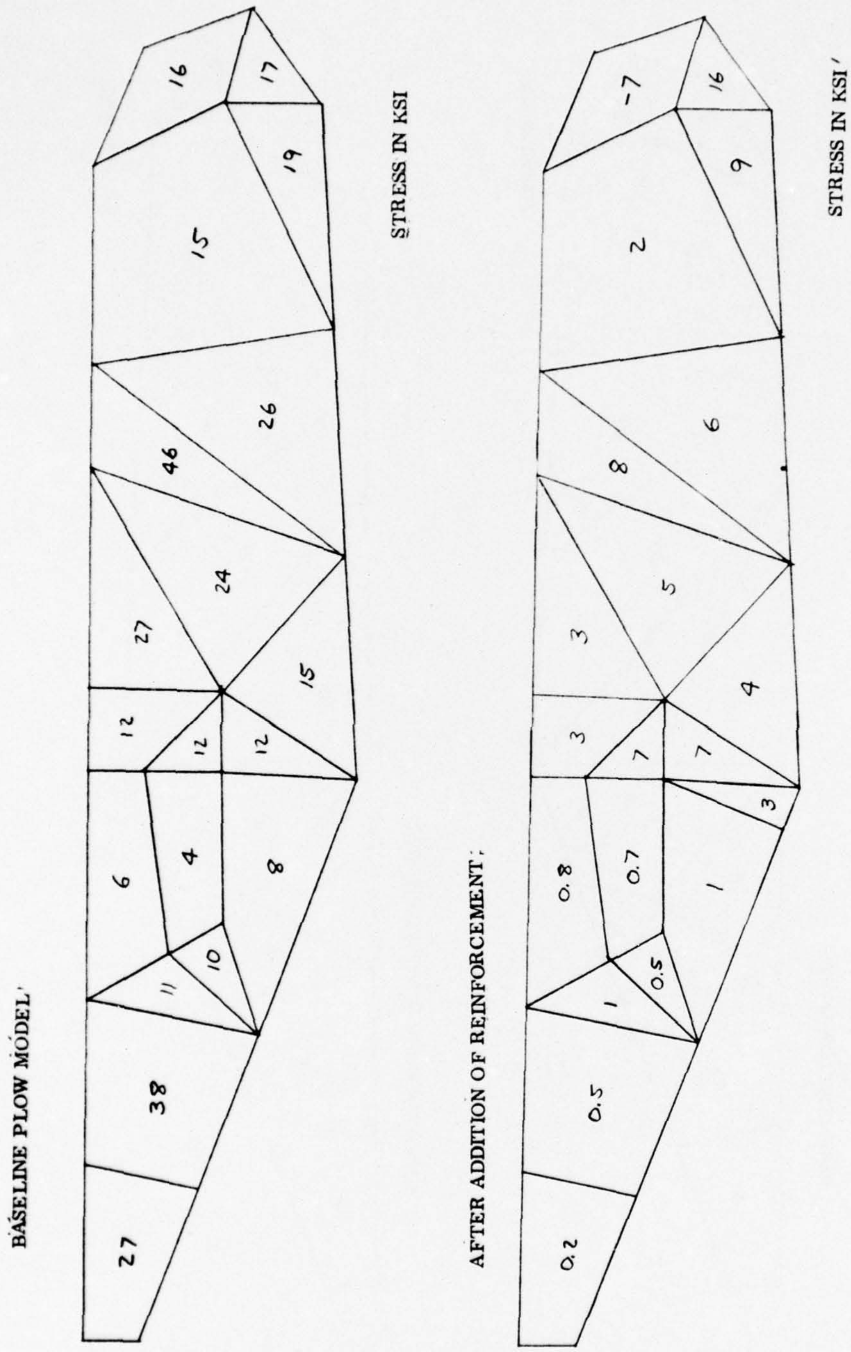


Figure D-9. Maximum Principal Stress in Mounting Bracket Inside Piece TD 135932-3

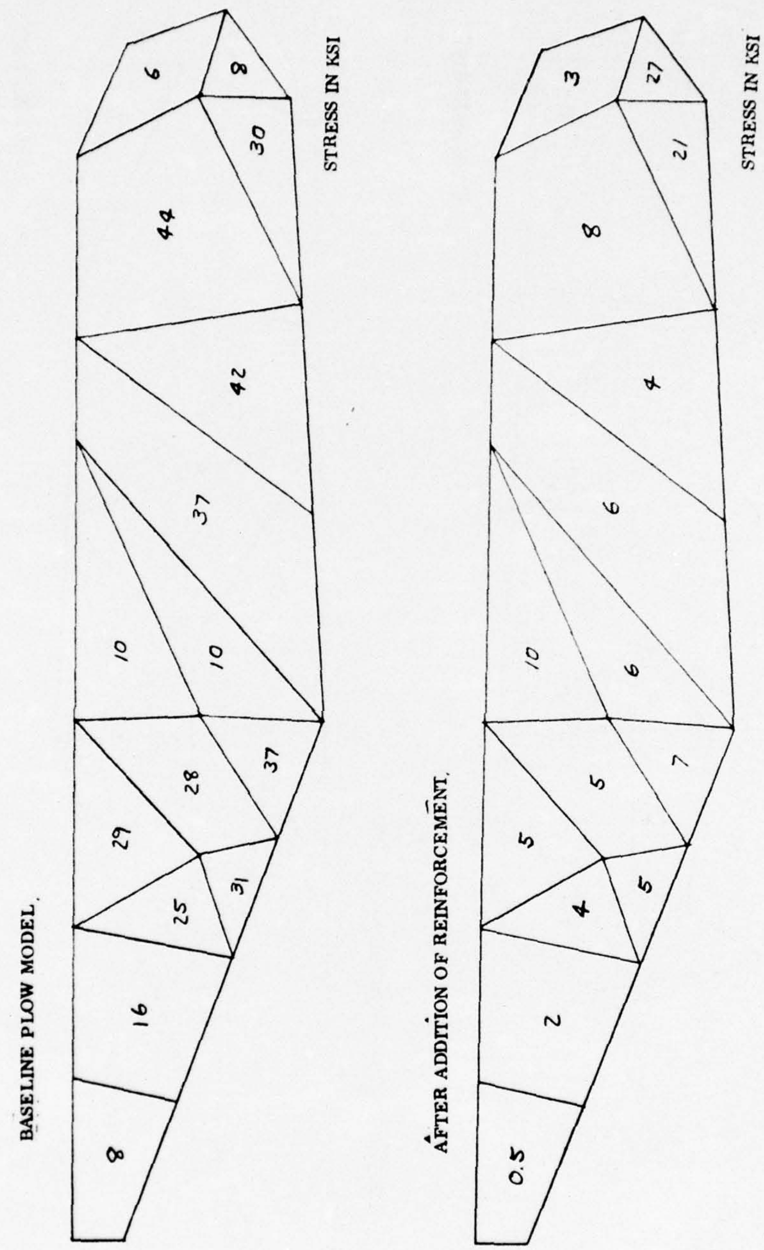


Figure D-10. Maximum Principal Stress in Mounting Bracket Outside Piece TD 135932-4

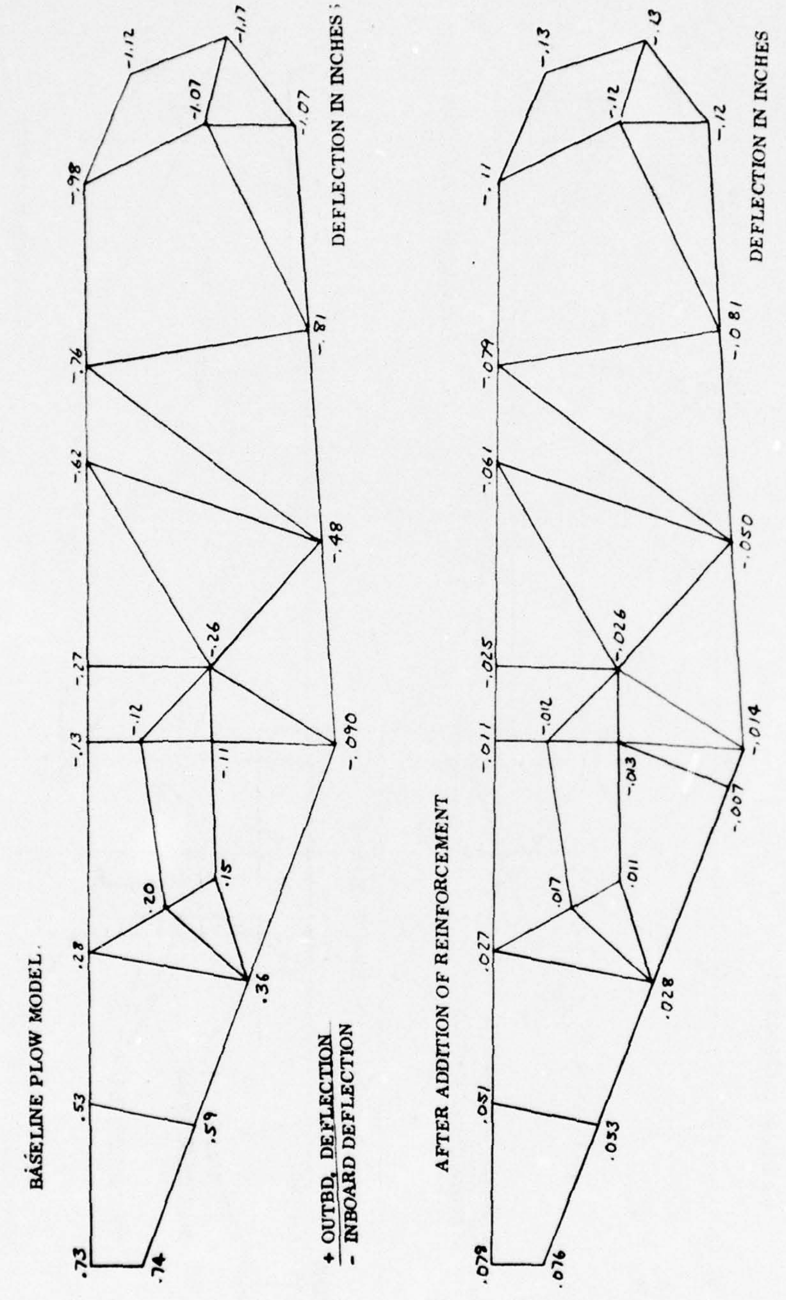


Figure D-11. Mounting Bracket Grid Point Deflection Normal to Plane of Plate - Inside Piece TD 135932-3

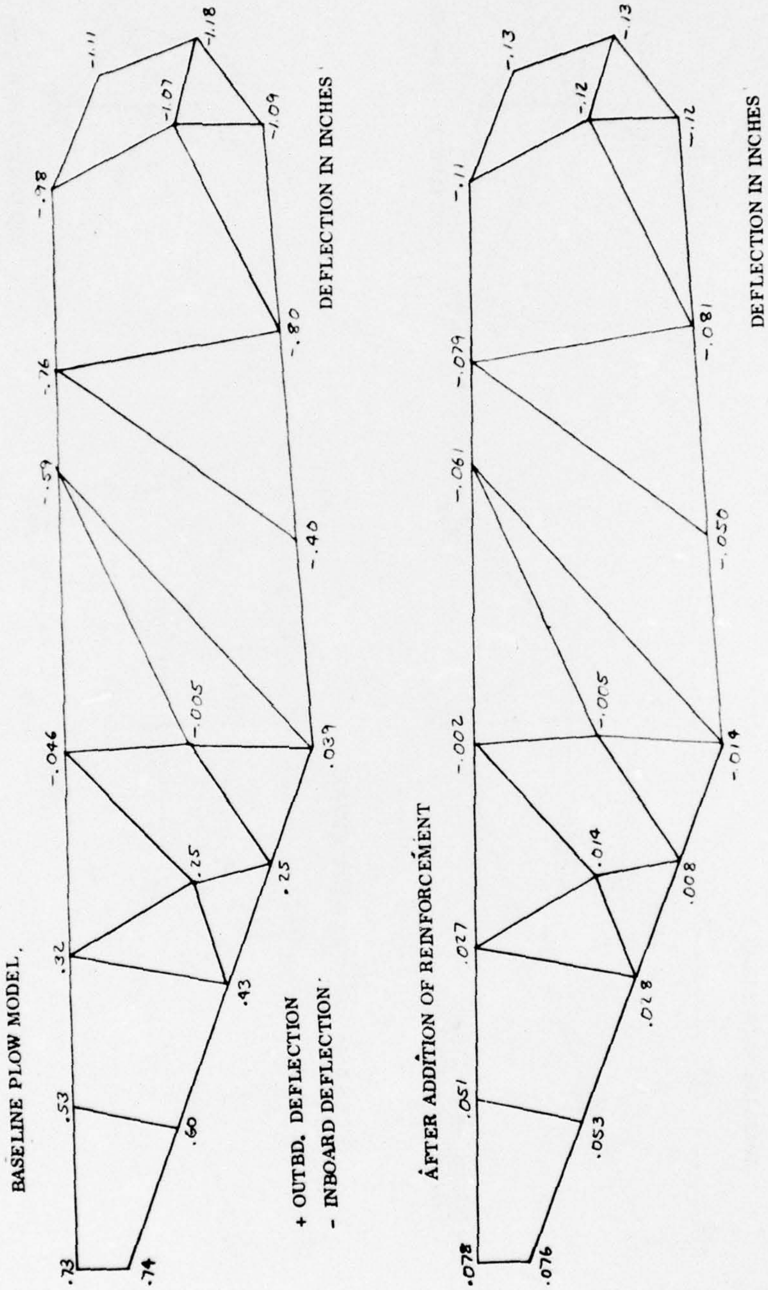


Figure D-12. Mounting Bracket Grid Point Deflection Normal to Plane of Plate - Outside Piece TD 135932-4

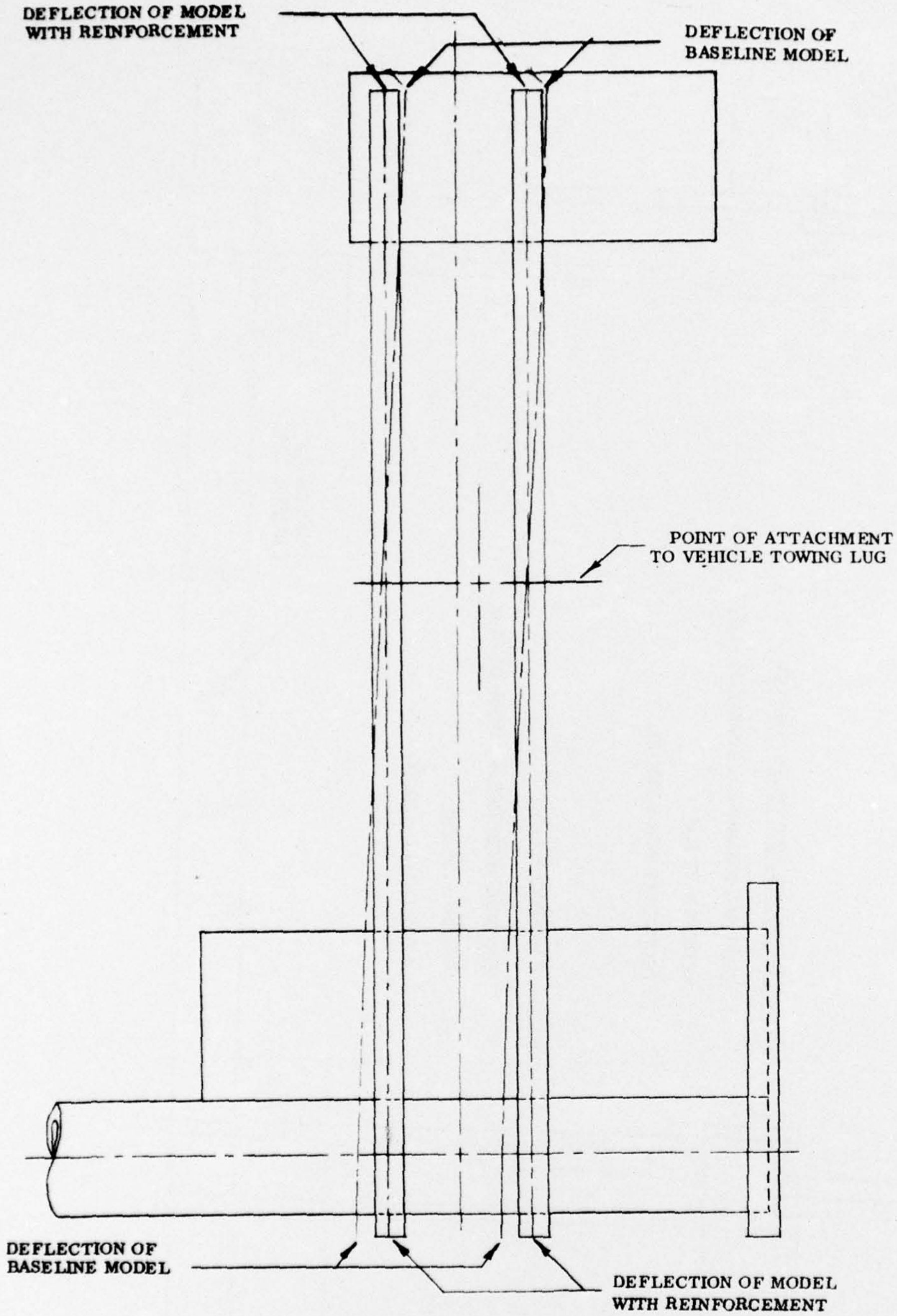


Figure D-13. Mounting Bracket Lateral Deflection Based on Nastran Results

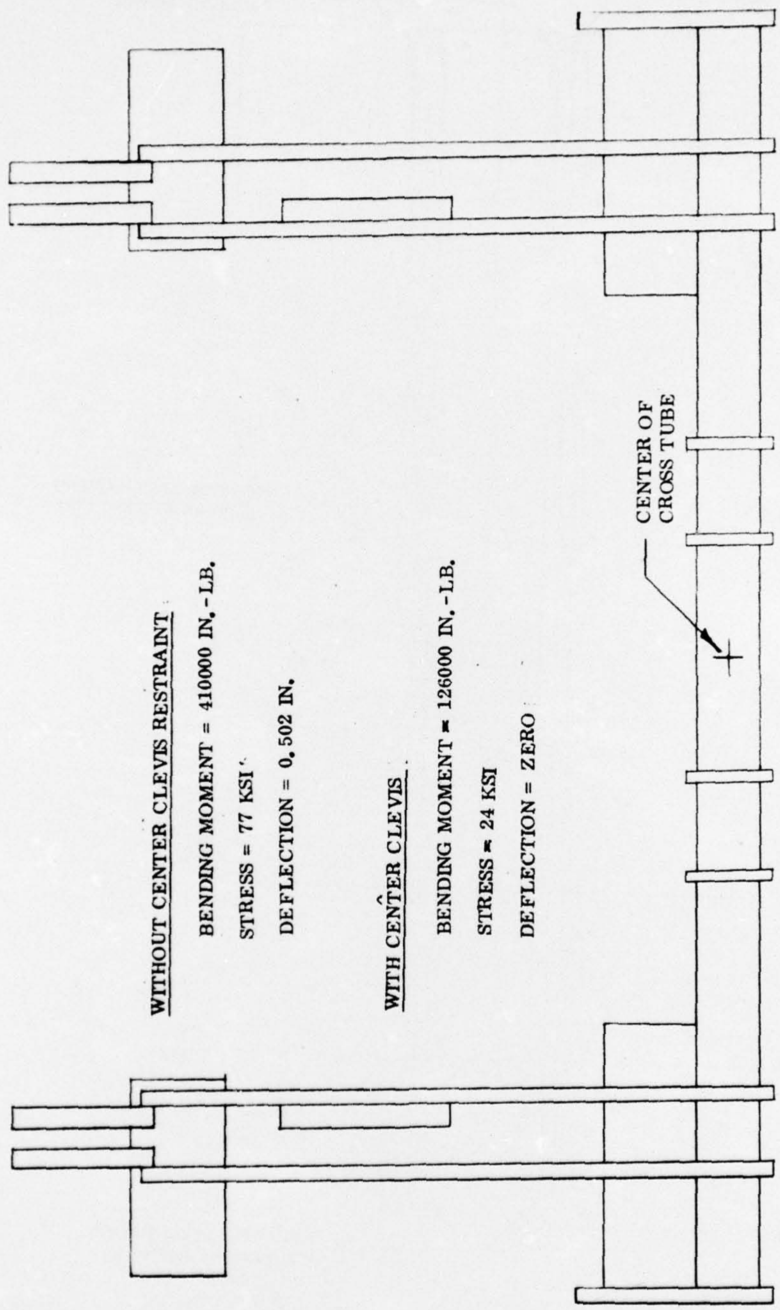


Figure D-14. Bending Stress at Center of Lateral Cross Tube

APPENDIX E

PRINCIPLE ELECTRONIC COMPONENT FUNCTIONS

To support an understanding of the E-MALS alignment procedure and permit easier troubleshooting, the following functions of the electronic components are provided. These components are referenced on the Mine Plow Board Assembly Schematic diagram, figure 16.

CR_1 - CR_2	Combines UP and DN signals for power relays and SCR power source.
CR_3 - CR_4	Combines AR_1 and AR_2 commands for operation of relay driver amplifier Q_{11}.
CR_5 - CR_6	Combines AR_1 and AR_2 commands for operation of relay driver amplifier Q_{11}.
CR_7	Trigger level for Q_3.
CR_8	Sets SCR_1 trigger voltage from C_1.
CR_9	Steering for amplifier Q_{11} (Partial Up Signal).
CR_{10}	Steering for partial DN signal. (To Q_8 isolates Q_1 & Q_3).
CR_{11}	Steering for partial DN signal.
CR_{12} - CR_{13}	Transient suppression for relay coils.
CR_{14} - CR_{15}	Isolates negative spikes on command signals.
CR_{16}	Isolates SCR_1 power from negative spikes.
CR_{17}-CR_{18}-CR_{19}	Isolates UP, and SCR_1 DN and overload light circuits from test voltage commands.
SCR_1	Provides trigger and hold of overload signal.
C_1	Filter for integrating AR_5 signal and delay for SCR_1.

PRINCIPLE ELECTRONIC COMPONENT FUNCTIONS (Continued)

C_2	Filter for SCR_1 power.
C_3	Filter for motor overload signal.
AR_1 - AR_2	Comparator for sensing UP and DN travel limits.
AR_3 - AR_4	Comparator for sensing UP and DN travel limits.
AR_5	Comparator for sensing overload of motor field voltage.
Q_1 - Q_7	Gates for selecting AR_1 and AR_2 signals.
Q_2	Overload clamp for position relay coil amplifier.
Q_3	Signal inverter for mode selection.
Q_8 - Q_9	Gates for selecting AR_3 and AR_4 signals.
Q_{10}	Position coil amplifier.
Q_{11}	Speed coil amplifier.
Q_{12} - Q_5	DN light driver and amplifier.
Q_6 - Q_4	UP light driver and amplifier.
R_1	Position sensing potentiometer bridge arm (TD 137751)
R_2-R_3-R_4-R_5	Fixed adjustable bridge arms.
Power Supply	Provides $\pm$ 15 volts for comparators.

APPENDIX F

E-MALS ALIGNMENT PROCEDURE

The attached alignment procedure and information must be considered preliminary and subject to change at this time.

The procedure in general is believed unnecessary (no adjusting should be required) and is provided for information only. The plow upper limit stop being the only adjustment subject to mechanical tolerances of castings, etc., might possibly be found to require resetting if mechanical adjustments are not adequate.

Reference DWG TD 137427 and TD 137751 (See Figure 16)

NOTE: One potentiometer turn of $R_{2-3-4-5}$ produces 1/4" travel of lead screw

CW rotation of $R_{2-3-4-5}$ produces DN travel

CCW rotation of $R_{2-3-4-5}$ produces UP travel

Step 1 Close switches S_1 and S_2 - move lever to right

Step 2 Position pot R_1 is set at electrical zero * using an ohmmeter and running motor - rotate lead screw to prevent travel

* Nos. 9 to 6 (GND) of TER block or pin 6 to GND of XPIA or XP1B

Step 3 Plow full UP position (R_2 wiper + 11.5 VDC) is set by adjusting pot R_2 as follows:

To move DN - turn CW

To move UP - turn CCW

Step 4 Plow full DN (R_5 wiper - 11.5 VDC) position is set by adjusting pot R_5 as follows:

To move DN - turn CW

To move UP - turn CCW

E-MALS ALIGNMENT PROCEDURE (Continued)

Step 5 Typical positions of the yoke with respect to the ball nut are:

DN = - 12 V)		1 1/16")	measured directly
UP = + 10.5V)	R_1 Pin 12 =	14 1/16")	between ball nut
Center = 0 V)		8 3/8")	and yoke back

Step 6 Measure plow running current - typical #2 average **

Basic plow with no weight added

UP current = 45 AMP @ 24V)
DN current = 12 AMP @ 25V) system volt ***

Basic plow with weight added (20% load)

UP current = 55 AMP @ 24V)
DN current = 12 AMP @ 25V) system volt ***

Basic plow with weight added (35% load)

UP current = 85 AMP @ 24V)
DN current = 12 AMP @ 25V) system volt ***

UP current = 80 AMP @ 27V)
DN current = 12 AMP @ 27V) system with charger added

Step 7 Check temperature rise of motor with thermo couple.

After 10 cycles (10" lift) with 530 lb.

$$T = 88° F \longrightarrow 107° F$$

** Average includes wide variations (15 AMP) due to:
- motor speed
- motor temp.
- system age
- angle of beam
- ball screw wobble
- overload setting

*** Battery only

E-MALS ALIGNMENT PROCEDURE (Continued)

Step 7 (cont'd) After 10 cycles (3" lift) with 1, 830 lb.

$$T = 98° F \longrightarrow 130° F$$

Step 8 Measure UP travel time for 22-1/2" travel

= 14 sec for basic plow alone ***

*** Battery only

Step 9 Adjust overload trip out setting.

NOTE: To increase current, turn R_7 CW

To decrease current, turn R_7 CCW

With reference to Figure G-1, the desired maximum overload current sustained is converted to R_7 voltage which is monitored at the test point on the plug in cord.

NOTE: Both plows have been set for 35% load or approximately 6V (R_7).

NOTE: Use of both plows simultaneously for periods of time in excess of two minutes may blow the main fuse or overheat the motors or both.

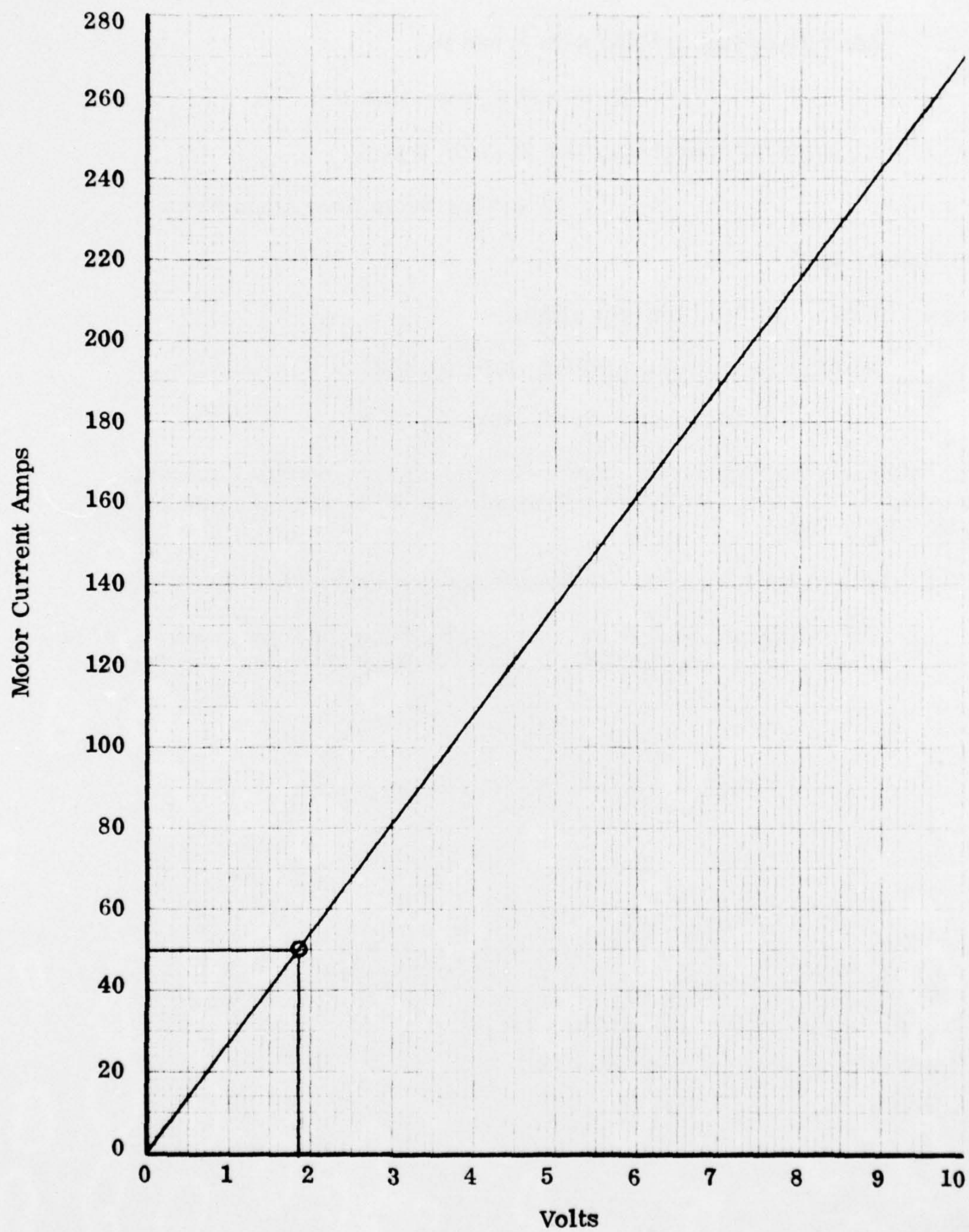


Figure F-1. Overload Adjustment R_7 at Test Points XP1A and XP1B

APPENDIX G

SUMMARY OF E-MALS CONTROL SYSTEM BASIC OPERATION

1. The "On" switch applies power to the control system. Control system power "ON" indicator will light. All indicators may be checked with the push to test switch at any time.

2. The "Up" and "Dn" switch directs the control system to raise or lower each plow. Switches may be left in "Up" or "Dn" or center "Off" positions at any time.

3. The "Up" and "Dn" lights will indicate when plow has reached "Up" or "Dn" position and will continue to indicate until selector switch position is changed or placed in center "Off" position.

4. "Plow moving" light indicates when plow is moving towards selected position.

5. "Overload" lights indicate when system has stalled and automatic protection circuit is operating. Position switch is reversed or turned off to reset protective circuits. Protective circuits will repeatedly turn system off until overload is removed.

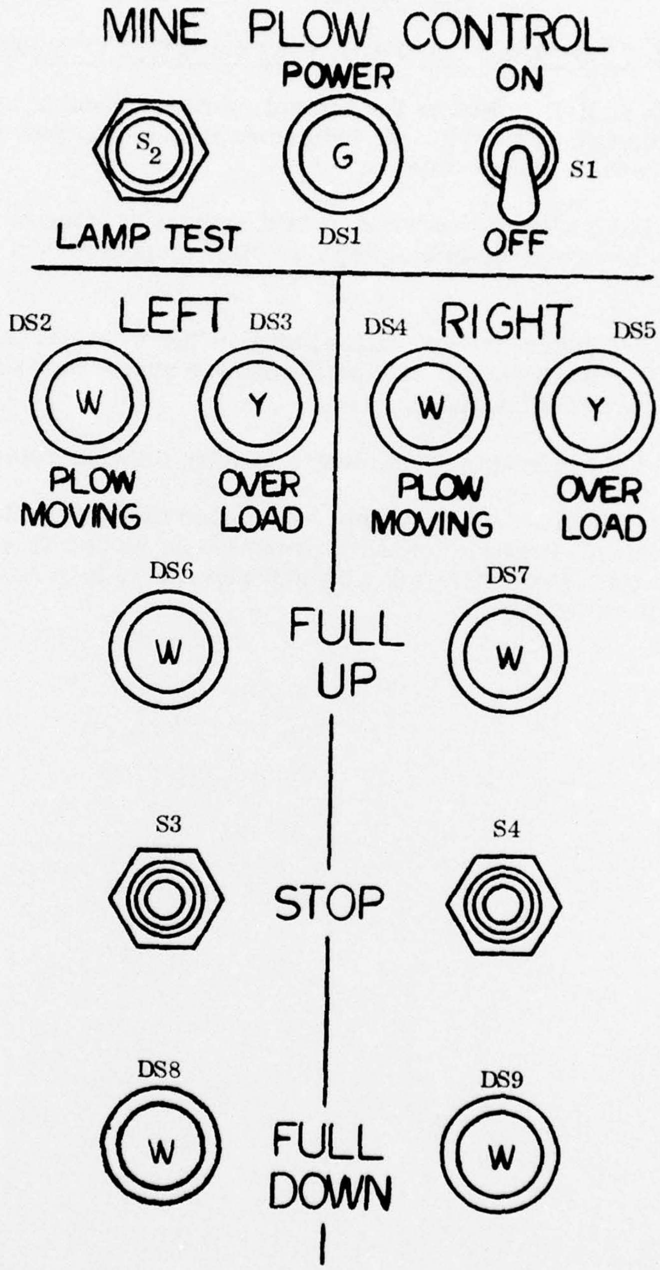


Figure G-1. Control Panel Pictorial

APPENDIX H

E-MALS MAINTENANCE

The delivered control system requires no periodic maintenance. The electric drive motor's brushes and commutator should be checked for wear conditions at every 100 hours of operation and as the opportunity arises when the actuator unit is removed from the pushbeam. Experienced personnel must make a judgment as to the repair maintenance required.

The actuator has two grease lube fittings accessible through the lift chain opening with the boot fitting removed when installed in the pushbeam. Any good grade of grease conforming to MIL-G-23827 (Shell Oil Co. Alvania EP#2 for example) can be used. One shot of grease should be applied as the opportunity for access arises. A full exchange of fresh grease should be applied (until the old grease is forced out through the seals) at least once per calendar year.

The gearbox takes about 1/3 of a quart of oil. Oil per MIL-C-2105C should be used, with a grade range of 75W to 90W. The gearbox oil level should be checked with a dipstick before every day's mission since leakage can occur at the oil seals. At least 3 inches of oil should show on the dipstick. Access for dipstick checking is by removing the circular cover plate on the top of the pushbeam. This exposes a hex recessed pipe plug in the gearbox cover. A 1/4-inch maximum thickness dipstick can be inserted at this point which will reach the bottom of the gearbox cavity. Care should be exercised to prevent any dirt or other foreign materials from entering the oil sump.

The oil life, if kept clean, is good for 6,000 hours; however, if the dipstick oil shows metallic wear particles, the oil should be changed at the first opportunity of actuator removal from the pushbeam. With the oil removed, the gear teeth should be inspected for wear by experienced personnel. Gear teeth showing abnormal wear patterns, flaking, or scoring marks should be replaced.

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13. ABSTRACT

This report documents the technical effort performed under contract DAAK02-72-C-0451, modifications P00012 through P00016, to resolve the major DT II Track Width Mine Plow EPR deficiencies and shortcomings assessed.

The changes addressed in this report include structural improvement along with replacement of the electro-hydraulic lift system with an electromechanical system capable of being quickly mounted to a vehicle. USAMERADCOM testing at Aberdeen Proving Grounds during the months of March and April 1976 of an M60A1 vehicle equipped with a Track Width Mine Plow System so modified, has demonstrated the feasibility of the improved system.

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